Investigation of the Operating Principle of Diffuser Augmented Hydrokinetic Turbines

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Abstract. Alternative sustainable technologies for energy production are currently receiving increased attention, one of them being hydrokinetic turbines. For some applications an augmentation of the turbine rotor through a diffuser is a reasonable option offering some advantages. In order to get a better understanding of diffuser augmented turbines an extensive literature study is performed also including relevant publications on ducted wind turbines. The published geometry and performance data are collected and reevaluated applying a new procedure, which references the turbine power on the diffuser outlet area instead of the rotor area. Additionally, geometries are generated, manually optimized and compared to the data gathered in literature. Dominant diffuser attributes are identified independent from the exact diffuser concept such as the shape of the diffuser close to the exit plane and the diffuser area ratio. By using the new evaluation approach a strong correlation between the area ratio and the overall turbine power coefficient is observed. Observations suggest that an increase of the exit area is only reasonable up to a certain limit. To a some extent, this constraint can be overcome through boundary layer control, e.g. by applying multi-stage diffuser concepts. The presented work gives a guideline for effective design of diffusers for hydrokinetic turbines and generally improves the understanding of the operating principle of diffuser augmented turbines.

1. Introduction

Driven by the transition towards sustainable energy production alternative technologies, like hydrokinetic turbines, receive increased attention. The application of hydrokinetic river turbines may give access to river sections where the construction of a conventional hydropower plant is not possible for ecological or environment reasons, exploiting unused energy potential. Hydrokinetic turbines use the kinetic energy contained in the flow of the working fluid. This may for example be driven by wind, tides or rivers. For common horizontal-axis single rotor turbines the Linear Momentum Actuator Disk Theory (LMADT) introduced by Lanchester and Betz provides good results for a first estimation of turbine power output under the assumption of an infinitely large flow field around the machine [1]. This 1D theory, which is based on the conservation of mass, energy and momentum, states that only a part of the available kinetic energy may be extracted from the flow, since a part of the energy is required to move away the fluid downstream of the rotor. The energy extraction leads to a deceleration of the flow, which is described by the rotor induction factor:

$$a_r = 1 - \frac{u_r}{u_\infty}$$
and the power gained by the rotor may be determined as:

\[
P_r = \frac{1}{2} \rho A_r u_\infty^3 c_{p,r} \quad (2)
\]

where \(u_\infty\) is the infinite undisturbed upstream flow velocity, \(u_r\) is the velocity in the rotor plane, \(A_r\) is the projected rotor area and \(c_{p,r}\) is the rotor power coefficient referenced on the projected rotor area. According to the theory the optimum operation point is reached for an induction factor of \(a_r = 1/3\), where a power coefficient of \(c_{p,r} = 16/27\) is theoretically possible. This means that at best around 60\% of the kinetic energy can be used by a turbine. The corresponding stream tube for the optimum operation point is schematically illustrated in the upper half of Figure 1.

![Figure 1: Schematic illustration of stream tubes induced by a single rotor turbine (top) and a diffuser augmented turbine (bottom)](image)

Figure 1: Schematic illustration of stream tubes induced by a single rotor turbine (top) and a diffuser augmented turbine (bottom)

Regarding Equation 2 it is obvious, that the main design parameter to increase the generated power is the projected rotor area \(A_r\). For some applications, e.g. for hydrokinetic river turbines, the rotor diameter and thus the area is limited by the flow depth. By equipping the rotor with a diffuser this limit may be circumvented as the projected area of such a turbine is not necessarily circular. Using a well-designed diffuser it is possible to achieve a slightly higher performance than using a stand-alone rotor with the same projected area. This is possible by reducing the tip vortex on the one hand and increasing the mass flow capacity on the other hand. A further advantage of this concept is that the rotor size may be decreased which allows a higher rotor speed and hence a reduction of the generator costs. Through the suppression of the tip vortex also the noise emission is reduced. Moreover, diffuser augmented turbines are less sensitive to inclined flow angles. But a diffuser is a large building structure with high forces acting on it. This requires an increased technical and financial effort. Furthermore, the concept is very sensitive from the hydrodynamical point of view, as the flow is not forced to go through the machine, but always has the opportunity to pass by. So a bad design can disturb the whole principle of operation of the diffuser. For this reason, important design parameters will be examined more closely in the following.

2. Background on Diffuser Augmented Turbines

Most of the early research on diffuser augmented turbines was performed in the field of wind turbines. Some of the first to work on this problem intensively were Lilley and Rainbird [2]. They tried to theoretically determine the augmentation of power obtained by using a diffuser. They were already able to identify important design parameters such as the diffuser expansion ratio and the external shape of the diffuser at the exit by means of their theoretical considerations.

Igra [3] was working on the same topic focusing more on the experimental investigation of different diffuser concepts. First he examined a wing-shaped diffuser. As for this design flow separations occurred, bleeding channels were introduced for boundary layer control (BLC). This
provided a significant improvement and finally led to the development of a new diffuser concept: a multi-stage diffuser consisting of two (or more) individual diffusers in a row. In a further step, he added a secondary wing-shaped diffuser at the rear end of his basic diffuser.

In the field of single-stage diffuser development Lewis [4] performed experiments on a straight-line contour diffuser around the same time. During the test series the diffuser area ratio was changed by varying its length at a fixed diffuser angle. The straight-line diffuser design was taken up by Gaden [5], who numerically investigated different diffuser geometries varying the area ratio and diffuser opening angle. Hansen [8] and later Shives [9] rather followed the example of Igar [3] for their diffuser concepts applying wing-shaped diffuser contours based on the NACA profile series. Both used numerical flow simulations for their investigations. Shives experimented with BLC through a gap between actuator disk and shroud as well. Also Grassmann [11] considered a large rotor gap of 35% the rotor radius. This allowed him to apply a large area ratio to his wing-shaped diffuser without observing flow separation at the diffuser walls from his CFD results. The diffuser presented by ten Hoopen [10] is also consisting of a circular airfoil. He was focusing on the experimental analysis of the benefits provided by vortex generators placed at the trailing edge of the diffuser. Also Ohya [6] considered the diffuser shape at the outlet to be important. He developed a straight-line diffuser which is characterized by a large flange fixed at the trailing edge. The size of this flange also leads to a large projected machine area and hence a large area ratio. In a later study he concentrated more on the testing of diffusers with more moderate area ratios and varying flange dimensions. Also the diffuser contour was changed to a curved more compact shape (wind-lens) [7]. Most of the single-stage diffuser concepts discussed here are schematically illustrated in Figure 2.

Parallel to the development of single-stage diffusers, the research in the field of multi-stage diffusers was pushed forward by the Grumman Aerospace Corporation, namely by Oman, Foreman and Gilbert [12]. They published analytical framework and experimental investigations on different variants of single-stage and multi-stage diffusers. In this framework, also the application of flange-like flaps at the diffuser outlet was tested. Besides the boundary layer control, these flaps was identified as an important design parameter. During the experiments the pressure drop at the rotor was simulated by screen meshes. The work at Grumman Aerospace on diffuser augmented wind turbines continued for a long period leading to many publications, such as [13], [17] and [18]. A group of New Zealand scientists picked up Grumman’s idea and together with Foreman developed the Vortec 7 turbine [19]. This diffuser was further improved by Phillips [14] through numerical and experimental testing. He finally presented three adapted geometry versions, of which the last one is characterized by a significant increase of the diffuser stages. At the same time Ruprecht and Reinhardt [16] developed a multi-stage diffuser for tidal
current energy application. Their diffuser concept was manually optimized using numerical flow simulation. To the best of the author’s knowledge this diffuser, including rotor and hub, was the first one to exceed the power of a single actuator disk with similar projected area. Also Hjorten [15] picked up the idea of boundary layer control through multiple diffuser stages and further developed the geometries presented in [7] and [14]. By introducing a multi-stage and multi-layer diffuser he could reach the highest power output in relation to the projected area found in this literature review. A summarizing overview of the multi-stage diffusers is provided in Figure 3.

It is generally agreed that augmentation of a single rotor through a diffuser leads to an increase of power. Furthermore, the scientific community agrees that with increasing area ratio, also the performance increases. This is obvious when considering the following statement of Lilley: “Hence the power output from a ducted windmill, where the external profile is cylindrical, is equivalent to that from an unshrouded windmill, having a disc area equal to the duct inlet (exit) area, ... “ [2]. So the exit area is a significant design parameter for an ideal diffuser augmented turbine in the same way as the projected rotor area is for a single rotor turbine. Therefore, it is surprising that most of the authors are referencing the power of their machine(s) on the rotor area by using the rotor power coefficient \( c_{p,r} \) or the augmentation factor \( r \). The diffuser outlet area may provide a more informative reference dimension, as it includes the efficiency of the diffuser to the examinations. This view is particularly relevant if the available space at the turbine site is limited, as is the case e.g. for rivers with small flow depths. For those applications designing a machine providing high power for a fixed diffuser outlet area is the main goal.

3. Alternative Evaluation Procedure

Based on the aforementioned reasons, an alternative overall power coefficient is defined, which is related to the diffuser outlet area \( A_d \):

\[
c_{p,d} = \frac{P_r}{\frac{1}{2} \rho A_d u_\infty^3}
\]  

(3)

This definition may also be found in [15]. The concrete meaning of this value can be described as a ratio of the achieved turbine power to the total kinetic energy available in the blocked/occupied cross area. So for an ideally shaped diffuser with no losses the maximum overall power coefficient should be in the range of the power coefficient defined by Betz. This concept is illustrated in the lower half of Figure 1. Also the deceleration of the flow has to be regarded in the context of the whole machine. Therefore, also the induction factor must be redefined as diffuser induction factor:

\[
a_d = 1 - \frac{u_r}{\gamma u_\infty}
\]  

(4)

where \( \gamma \) is the ratio between the diffuser exit area \( A_d \) and the rotor area \( A_r \). In this work the diffuser exit area is defined as the total projected area and thus also includes the projected area of flanges or brims, if existing.

The evaluation of the diffuser geometries designed within this work is carried out according to the presented concept. Accordingly, the geometry and performance data provided by the literature are reinterpreted. At this point a certain inconsistency in the gathered data must be discussed. For example in some cases geometric diffuser attributes like the diffuser outlet area were not clearly stated in the publications and had to be recalculated or measured from sketches. Furthermore, all the data were collected using very different investigation and evaluation methods. Some of the investigations were performed based on experiments, whereas other authors used numerical flow simulations. Also within the two methods there are large differences in methodology, e.g in the treatment of the rotor or the consideration of the hub. In
some cases the authors used rotor models like a screen mesh or the actuator disk model, which provides the available power in the rotor plane. Others like [4], [7], [10] and [11] considered the runner blade geometry and specified the performance based on the shaft power, which is lower due to the blade efficiency. Nevertheless, the collected data can give a good overview over general correlation, as this work is not about to identify the best diffuser design.

4. Simulation Methodology
All the geometries designed within this work are investigated using 3D numerical flow simulations. The different diffuser geometries are placed in a very large domain in order to avoid interactions with the domain boundaries, as exemplarily shown in Figure 4. Since all examined geometries are rotationally symmetrical, it is reasonable to reduce the solution domain to a section of 10° of the full geometry in order to minimize the computational effort. This sectioned solution domain is discretized through a block structured mesh for each diffuser geometry. An exemplary mesh resolving a multi-stage diffuser is shown in Figure 4.

Diffuser augmented turbines designed at the Institute of Fluid Mechanics and Hydraulic Machinery of the University of Stuttgart (IHS) usually include guide vanes and runner blades. For an ideal geometry, this leads to a swirl-free flow in the diffuser. Therefore, the guide vanes and runner blades are not directly resolved but modeled using a source term similar to an actuator disk in this work. A constant quadratic loss coefficient is applied to model the pressure drop induced by the energy extraction through the runner. In the course of this simplification also the hub geometry is neglected. Using this rotor model is a common approach, not only in the numerical [5, 8, 9], but also in the experimental investigation of kinetic turbines where screen meshes or gauze are used to simulate different disk loading coefficients [3, 4, 12, 13, 14]. As now the swirl is neglected and all the investigated operation points are generally chosen close to the optimum where no strongly disturbed flow is expected, the k-ω-SST-Model is applied for turbulence modelling and a steady state flow situation is assumed. At this point it should be noted that numerical simulations of single actuator disks considering turbulence provides a slightly higher power coefficient then predicted by the LMADT [5, 20].

5. Analysis of Sensitive Diffuser Attributes
Based on the prediction of [2], the shape of the diffuser at the exit and the diffuser expansion ratio (area ratio) are examined more closely in this work. Furthermore, the presented evaluation distinguishes between single- and multi-stage (BLC) concepts. For this purpose, geometries are designed and examined in detail. In addition, the data collected from literature are evaluated according to the approach described above and all gathered data are compared and brought into context.

5.1. Geometric Shape at the Diffuser Exit
For the investigation of the significance of the diffuser shape at the diffuser outlet, a basic diffuser geometry \( V_0 \) is defined. As illustrated in Figure 1, the inner contour of this diffuser is generated
from the stream tube upstream of an ideal actuator disk at optimum operation conditions (CFD results). This again implies that the diffuser outlet area is, comparable to the rotor area for a single rotor machine, the relevant area of action. The inner contour is thickened and provided with an elliptical leading edge and the variable wall thickness results in a pointed trailing edge. Due to the nature of the LMADT the area ratio of this basic diffuser is $\gamma = 1.5$. Derived from design guidelines for a stable diffuser flow in conventional hydropower plants an overall opening angle of less then $8^\circ$ is applied for the basic diffuser version [16]. The resulting design looks quite similar to the ones presented by [8] and [9] and with $c_{p,d} = 0.555$ provides a diffuser power coefficient of similar dimensions. In order to optimize this basic diffuser design and increase the power coefficient of the machine, the overall length of the diffuser is varied. This leads to different exit angles at the rear edge of the diffuser for a constant diffuser area ratio of $\gamma = 1.5$. The different geometry variants are presented in Figure 5a.

![Diagram of different geometries](image)

(a) Overview of different geometries

![Diagram of diffuser power curves](image)

(b) Diffuser power curves

Figure 5: Influence of the flow angle at the diffuser outlet

As shown in Figure 5b the basic diffuser V0 provides the lowest power output of the investigated diffusers. With increasing outlet angle the diffuser performance increases until reaching a maximum of $c_{p,d} = 0.633$ for the diffuser version V3. This power coefficient initially appears unrealistic, as it is well above the Betz-Limit. However, it must be noted that the flow may be redirected sharper through the fix contour of the diffuser than by mere deceleration at a single rotor. Consequently, a larger opening of the downstream stream tube can be achieved for the optimum operation point increasing the mass flow capacity and the upstream catchment area of the machine. With increasing flow angle, also the size of the flow separation zone at the diffuser outlet increases (see Figure 5a). This results in a decrease of wake pressure and hence of available head as already observed by [13]. When a certain size of the flow separation zone is exceeded, the positive effects are overridden by the negative effect of the blockage of the effective exit area. This may be observed for geometry version V4 where the overall turbine performance decreases significantly. In general, the results observed here point out to the principle of a design concept known as spoiler or flange effect, which is well known in relevant literature [6, 7, 12, 13, 15]. An opposite effect can be observed for a cylindrical diffuser without any opening ($\gamma = 1$), since here the streamlines follow the geometry and hardly widen. This leads to a very poor machine performance, as observed by [5] (see also Figure 7a).
5.2. Diffuser Area Ratio

As already mentioned the well performing diffuser developed above has an area ratio of $\gamma = 1.5$. In order to now examine the influence of the area ratio on the overall machine performance two further diffusers with the area ratio of $\gamma = 2.25$ and $\gamma = 4.0$ are designed. The geometries are manually optimized, testing and comparing around ten geometry versions for each area ratio. During the optimization process it became apparent that it is beneficial for such geometric conditions to use a stepped spoiler, detached from the actual diffuser contour through shape edge. Through this design flow separations can be better kept out of the inner diffuser flow. The resulting geometries are summarized in Figure 6a.

![Figure 6: Overview of best diffuser geometries designed in this work](image)

As described in Section 3 the results of the numerical investigation of those geometries are compared to geometry and reinterpreted performance data provided by the literature. The results are summarized in Figure 7a and show a clear correlation between the investigated variables. It may be observed that for increasing diffuser area ratios the overall power coefficient $c_{p,d}$ is rapidly decreasing. The reason for this behavior may be explained regarding the flow fields at optimum operation in Figure 7b. In order to reach an optimum deceleration at the diffuser outlet, very high velocities are required in the rotor area in case of large area ratios. Although a conservative overall diffuser opening angle of less than $8^\circ$ is maintained here (leading to the large diffuser lengths) the fast flow in the rotor plane does not follow the diffuser contour very well.

![Figure 7: Influence of area ratio on overall turbine performance for single stage diffusers](image)
tube, ending in a very large reservoir. The resulting rotor velocity distribution from the free flow simulation is used as inlet boundary condition. The simulation setup is illustrated in Figure 8a.

![Simulation concept](a) Simulation concept

![Axial velocity fields](b) Results: axial velocity fields

Figure 8: Comparison of flow field in diffuser with $\gamma = 4$ in free flow and in large tank

Regarding the resulting flow fields in Figure 8a it may be observed that the backwater zones are independent from the free flow around the diffuser and also occur for a conventional draft tube of same shape. But in conventional hydropower applications those separation do not significantly influence the mass flow rate and thus the turbine power. In case of a hydrokinetic turbine, where the flow has the possibility to bypass the machine, the separations are blocking the effective area and decrease the mass flow capacity of the diffuser. Therefore, an optimum induction factor of $a_d \approx 1/3$ and hence a good overall power coefficient may not be reached for single-stage diffusers with large area ratios.

### 5.3. Boundary Layer Control

Literature suggests that the performance of diffuser augmented turbines may be improved by applying any form of boundary layer control. This may be implemented for example by considering a large rotor gap, bleeding, diffuser vents or a multi-stage diffusers concept.

![Diffuser power coefficient](a) Diffuser power coefficient

![Axial velocity flow field for different area ratios](b) Axial velocity flow field for different area ratios

Figure 9: Influence of area ratio on overall turbine performance including diffusers with BLC

In order to determine if the correlation between overall power coefficient and area ratio is influenced by boundary layer control the diffuser presented by Ruprecht [16] is reinvestigated,
not considering the runner blades and hub. Besides this geometry having an area ratio of $\gamma = 2.25$ two further diffusers based on Ruprecht’s concept are designed, having an area ratio of $\gamma = 1.5$ and $\gamma = 4.0$ respectively. Those geometries are illustrated in Figure 6b.

The results from literature in combination with the results generated in this work are presented in Figure 9a. The results for single-stage diffusers presented in Figure 7a are marked as gray points in the background to allow a direct comparison. It may be observed that for most of the geometries better results may be achieved by diffusers using boundary layer control. This proves that the concept generally may improve the diffuser design. The resulting flow fields displayed in Figure 9b show a fairly homogenous velocity distribution in every cross section inside the diffuser independent from the area ratio. The flow separations observed for single-stage diffusers are pushed downstream of the diffuser or are even totally suppressed.

Furthermore, the good results achieved by the geometry with $\gamma = 4.0$ indicate that the limitation of the overall power coefficient through the area ratio discussed in the previous section may also be improved through the boundary layer control. But this value is an exception in the gathered results. It still has to be examined in detail whether the observed trend continues for larger area ratios. A further possible reason for achieving a value that high is the neglect of the hub geometry in this case.

One reason for the improvement observed when applying boundary layer control is avoiding the flow separations and hence increasing the mass flow capacity of the diffuser. This is reflected in a lower diffuser induction factor, as shown in Figure 10. Especially the superior designs, like Hjorten’s [15] and Ruprecht’s [16] geometry, are characterized by this property. The increased mass flow capacity is also visible in Figure 11, showing a direct comparison of the flow situation at the same diffusers with and without boundary layer control for an area ratio of $\gamma = 1.5$ and $\gamma = 2.25$. It may be observed that the velocity in the rotor plane may be significantly increased through the boundary layer control, as the flow follows the contour and leads to a well opening downstream stream tube.

6. Conclusion and Outlook
A systematic analysis of different geometric attributes of diffuser augmented hydrokinetic turbines is presented. The work is based on the investigations of own diffuser designs compared to data gathered in an extensive literature study. In contrast to previous work, the evaluation
was based on characteristic diffuser quantities referenced on the diffuser outlet area instead of the rotor area. This approach allowed to identify a clear correlation between the diffuser area ratio and the overall machine power coefficient. Due to flow separations at the diffuser walls close to the exit, occurring for high area ratios, a good operation point for the free flow situation may not be reached and the overall power coefficient decreases dramatically. This situation may be improved by boundary layer control, where higher area ratios are possible without observing a dramatral decrease in overall power coefficient. In general, it becomes clear that for an optimum design a balance between an efficient diffuser and a large aspect ratio (small rotor) is necessary - especially for applications with limited available space.

For a detailed determination of the correlation between area ratio and overall power coefficient it might be useful to perform an automatic optimization of the diffuser geometry using multiple targets in order to get a pareto front defining the correlation between an optimum overall power coefficient and an optimum rotor power coefficient. Furthermore, in this work the hub geometry was neglected as well as the detailed rotor geometry. For future work it might be interesting to investigate their influence on the results as already indicated in Section 5.2. Finally, a detailed investigation of non-circular diffusers is desirable in order to assess to what extent the correlations observed in this work can be transferred.

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