Analysis, Design and Validation of a Vaned Diffuser for Improved Fish Friendliness

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The primary cause of mechanical-related fish injury and mortality in turbomachinery is blade strike. Fish contained in the flow may strike with the rotor blades and the fixed diffuser vanes, the latter being a non-negligible factor causing fish damage in the pump system. In this study, an experiment-based correlation of fish mutilation ratio acts as critical strike velocity. The relation between strike damage in a vaned diffuser and the theoretical pump head is presented as a function of specific speed. As an example, a vaned diffuser is designed for a single-bladed, mixed-flow impeller with the purpose of improving fish friendliness. This pump can be scaled to operate with a head up to 14 m at peak efficiency, without fish damage in the diffuser. Subsequently, experiments are conducted to show the retained pump performance as well as the great improvement of fish friendliness. [DOI: 10.1115/1.4052735]

Introduction

In order to develop so-called environmentally friendly hydropower turbines, the U.S. Department of Energy (DOE), Electric Power Research Institute (EPRI), and the Hydropower Research Foundation established the Advanced Hydropower Turbine System Program in 1994. The main goal of this program is to reduce or eliminate adverse environmental effects, among which the highest priority problem is the injury and mortality to fish as they pass through turbines [1]. Over the years, many investigations on fish injury and mortality indicate that the main mechanisms are rapid pressure change, shear stress, cavitation, and mechanical injury [2–4]. After identifying the damage mechanisms to fish, biological criteria were studied and established by several studies [5–10], which could make turbines more fish-friendly when incorporated in the new designs [4].

In a later study by Cook et al. [11], computational fluid dynamics (CFD) analyses at turbine best efficiency point (BEP) indicate that pressure change and shear stress exceed the fish survival criteria only in a small region near the blade leading edges and neighboring surfaces. It means that the duration for fish passing through the hazardous volumes is fairly short and should not cause significant fish mortality. Based on CFD simulations, Van Esch [12] also concluded that blade strike is the primary cause of damage to fish passing through pumping stations operating at pressure heads up to about 8 m. Von Raben [13] was the first to propose a simple blade strike model for predicting the probability of blade strike in Kaplan turbines. It was later expanded to Francis turbines by Monten [14] and Solomon [15]. Turnpenny et al. [7] examined the collision between fish and different blade profiles to establish the correlations for low-head, axial-flow tidal turbines. These statistical models were modified in Turnpenny's later studies. It was found that the probability of strike injury was highly dependent on fish size, turbine type, runner diameter and rotation rate, the number of blades, and operating load [8], resulting in the definition of the so-called mutilation ratio. Ploskey and Carlson [16] used deterministic and stochastic models to predict the blade strike probability in turbines and concluded that the orientation of fish and the location at which fish entered were the two most significant factors and uncertainties in the predictive models. Additionally, some models were developed to assess fish damage in open-type, horizontal-axis tidal turbines, such as the encounter rate model (ERM), the exposure time population model, and the collision risk model [17]. Although the resulting models are very similar to the blade strike model of Von Raben, some practical ideas can be extracted (e.g., avoidance rate and animal density) which are likely to improve the accuracy of the mathematical models. In all these mathematical models, fish are assumed to enter the runner along with the flow without active and unpredictable behaviors, leading to a wide confidence interval in the estimations of strike probability [12]. Recently, a series of further studies were implemented by EPRI to evaluate the influences of leading-edge thickness, cross section shape, and strike velocity on fish survival. The ratio of fish length to blade thickness was found to be an important parameter causing fish damage and also introduced to normalize the equations [18,19]. EPRI's results graphically show a strike survival rate with respect to the ratio of fish length to leading-edge thickness in a range of strike velocities. One of the recommended biocriteria for hydrokinetic turbine design is to keep the strike velocity below 4.8 m/s to prevent fish damage [19].

The same design principle can be used to avert or mitigate fish damage in pumping stations [20]. According to monitoring data of more than 20 pumping stations assembled with a different types of pumps, axial flow pumps turn out to be the least fish-friendly [21]. However, since high-specific speed pumps are of smaller size and thus the cheaper alternative, end-users are inclined to select high-specific speed, axial-flow pumps for their pumping stations. As a result, the development of fish-friendly, axial-flow pumps is drawing more and more attention. Van Esch [12,21] predicted fish mortality in the pumps using a blade strike model and

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Contributed by the Fluids Engineering Division of ASME for publication in the Journal of Fluids Engineering. Manuscript received August 23, 2021; final manuscript received October 7, 2021; published online January 12, 2022. Assoc. Editor: Chunlei Liang.

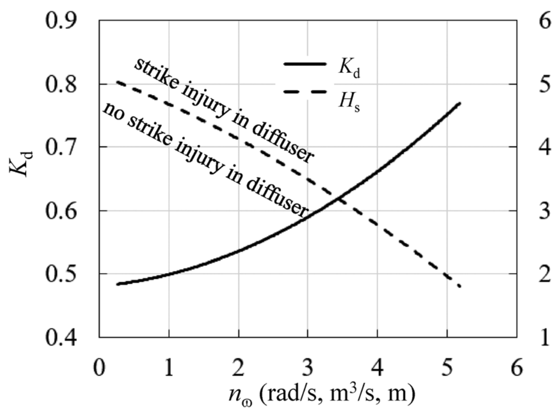

Fig. 1 The range of pump head without strike injury in the diffuser, as a function of specific speed n_ω

obtained good agreement with monitoring data of pumping stations. The model uses a correlation of mutilation ratio data by EPRI. Krakers et al. [22] estimated the fish survival rate for different components of a pump and suggested that the effective strike velocity should be the component perpendicular to the leading edge of the blade. It sheds some light on the reason why some fish-friendly pumps with a curved leading edge show a reduced risk of strike mutilation: it alters the included angle between relative velocity and leading-edge curvature, and reduces the strike velocity and thus the risk of damage. In hydrokinetic turbines, guide vanes are not regarded as a threat to fish because of the relatively low approach velocities (typically lower than 3.3 m/s) [19]. However, the situation is different in pump systems where the flow and fish are accelerated after passing through the impeller and hit the diffuser vanes at a higher speed. Besides, diffusers tend to have a relatively large number of vanes to match the impeller. For example, commonly used blade/vane number combinations are 7/12, 5/8, and 3/5. As a result, the vaned diffuser is a non-negligible component causing fish strike damage in a pump system.

Impeller (rotor) and diffuser (stator) are the two most important parts in a pump, while many studies only pay attention to the rotating rotor [23–27]. In this paper, however, the focus is on investigating the effect of diffuser vanes in a pump system on fish mortality, caused by the strike of fish with diffuser vane leading edges. Based on a conventional pump design method, the correlation between the head of the pump and the strike velocity with the diffuser vanes is obtained as a function of the specific speed. A design method for fish-friendly vaned diffusers with a reduced strike damage rate is presented and applied to an existing single-bladed, mixed-flow pump. The hydraulic performance and damage to fish for the original pump and the modified pump with vaned diffuser are compared and discussed using experimental data.

Strike Damage in a Vaned Diffuser

A strike is usually defined as a collision between a fish and the leading edge of rotating blades or fixed guide vanes [2]. In the blade strike model, the probability of a strike and the strike mutilation ratio are the two factors that govern fish mortality. Based on laboratory data of EPRI for rainbow trout, a regression equation of strike mutilation ratio is given by Van Esch [21] defined as Eq. (1). It indicates that a strike velocity below 4.8 m/s does not lead to damage for trout regardless of the blade thickness or fish length. This threshold is later used in Fig. 1

$$f_m = \left[a \ln\left(\frac{L_f}{d}\right) + b\right](v_s - 4.8) \tag{1}$$

Table 1 Regression analysis for *a* and *b*

L_f/d	*a*	*b*
0∼2	0.0531	0.0202
2∼10	0.0829	0.0021
10∼25	0.0327	0.1146

where f_m is the strike mutilation ratio, L_f is the fish length, d is the leading edge thickness of a blade or diffuser vane, v_s is the strike velocity, and coefficients a and b are given in Table 1.

In a well-known pump design method presented by Stepanoff [28], the values of some design parameters are recommended to avoid cavitation and to maximize efficiency, which consequently works as a design guideline for pump manufacturers. Based on this method, a correlation between the strike velocity at the diffuser vane leading edge and the theoretical pump head can be deduced.

Assuming that fish are rigid objects that move passively with the flow along the streamlines and assuming that the leading edge of diffuser vanes is normal to the streamlines, the strike velocity equals the absolute velocity of the flow at the leading edge, written as

$$v_3 = \sqrt{v_{u3}^2 + v_{m3}^2} \tag{2}$$

with v_3 the strike velocity at diffuser entrance, and v_{u3} and v_{m3} the components of v_3 in circumferential and meridional directions, respectively.

Based on the conservation of mass and angular momentum, the velocity components v_{u3} and v_{m3} at the diffuser leading edge can be associated with the velocities at the impeller exit and rewritten as

$$v_{u3} = \frac{v_{u2}}{1 + s \sin\varepsilon} \tag{3}$$

$$v_{m3} = \frac{v_{m2}}{(1 + s \sin\varepsilon)^2} \tag{4}$$

with v_{u2} and v_{m2} the circumferential and meridional components of the absolute velocity at the impeller exit, respectively, ε the angle between outflow direction and rotating axis as shown in Fig. 2, and s the gap ratio defined as the gap length L_g over the geometric average of the diameter D_{2m} at impeller exit, calculated in Eq. (7).

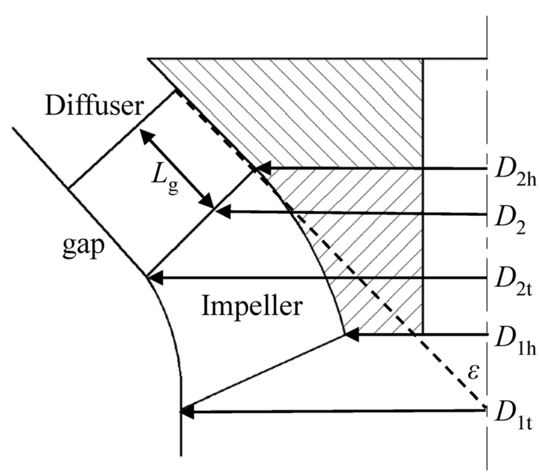

Fig. 2 Sectional drawing of a pump. The subscripts 1, 2, t, and h denote the inlet, outlet, tip, and hub of the impeller, respectively.

Following Euler's turbine equation and assuming uniform angular momentum, the value of v_{u2} has its maximum at the impeller hub and can be written as

$$v_{u2h} = \frac{gH}{u_{2t}h_2} \tag{5}$$

with g the gravitational acceleration, H the pump head, u_{2t} the tip velocity at impeller exit, and h_2 the hub ratio defined as the ratio of impeller exit hub diameter to tip diameter.

Correspondingly, strike velocity v_3 reaches a maximum at the diffuser hub, written as

$$v_{3h} = \sqrt{\frac{v_{u2h}^2}{(1+s\sin\varepsilon)^2} + \frac{v_{m2}^2}{(1+s\sin\varepsilon)^4}} \tag{6}$$

Introducing the geometric average of the diameter at the impeller exit, defined as

$$D_{2m} = \sqrt{\frac{D_{2h}^2 + D_{2t}^2}{2}} \tag{7}$$

where D_{2h} and D_{2t} are the hubs and tip diameters at impeller exit, respectively, the hub ratio h_2 can be written as

$$h_2 = \sqrt{2\left(\frac{D_{2m}}{D_{2t}}\right)^2 - 1} \tag{8}$$

In Stepanoff's design method, a number of dimensionless coefficients are related to a specific speed, such as K_{m2}, K_u, D_{1t}/D_{2m}, and D_{1t}/D_{2t}. The first two are defined below and D_{1t} denotes the impeller inlet tip diameter, as shown in Fig. 2

$$K_{m2} = \frac{v_{m2}}{\sqrt{2gH}} \tag{9}$$

$$K_u = \frac{u_{2t}}{\sqrt{2gH}} \tag{10}$$

Combining the equations above, a new dimensionless coefficient K_d, representing the ratio of the maximum absolute velocity of flow (or strike velocity) at the diffuser inlet to the velocity based on the theoretical pump head is defined as

$$K_d = \frac{v_{3h}}{\sqrt{2gH}} = \sqrt{\frac{1}{4K_u^2h_2^2(1+s\sin\varepsilon)^2} + \frac{K_{m2}^2}{(1+s\sin\varepsilon)^4}} \tag{11}$$

To avoid excessive pressure pulsations and hydraulic excitation forces, the gap ratio is recommended to increase from roughly 0.02 for a low specific speed pump to 0.15 for an axial flow pump, as a linear function of specific speed [29]. Besides, the angle ε normally decreases from 90 deg for a centrifugal pump to 0 deg for an axial flow pump. Then the nondimensional coefficient K_d will vary only with a specific speed and exhibit an increasing trend, as shown in Fig. 1. For a certain specific speed pump, the impeller tip speed can be selected to provide the required head. The higher the requested head, the higher the strike velocity at the diffuser will be.

In case the strike velocity is set to 4.8 m/s as the limiting value to prevent fish mortality, the corresponding upper limit of the pump head H_s is obtained. This means that a strike by diffuser vanes is not fatal to fish if the pump runs with a head lower than the specified head H_s. In Fig. 1, a wider range of heads is available for a low specific speed pump, indicating that pumps running at higher specific speeds are more likely to cause fish strike damage in the vaned diffuser. Notice that even if no strike mortality occurs to fish in the vaned diffuser, fish mutilation may still occur in the impeller [21].

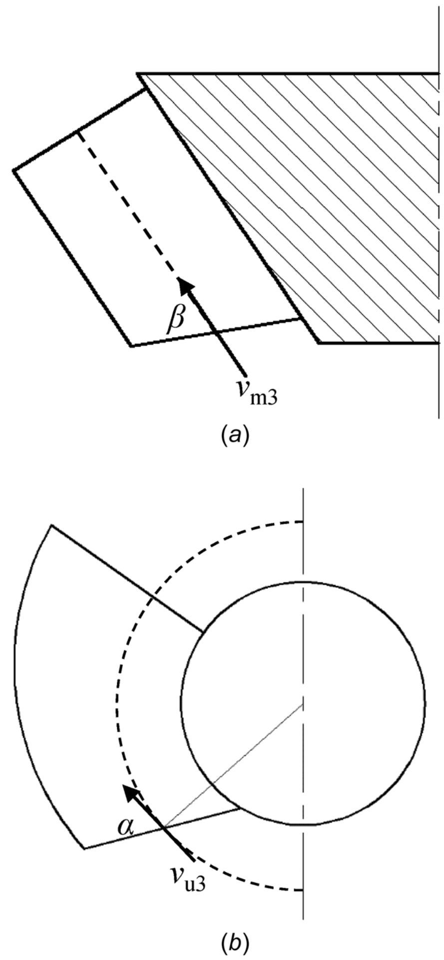

Fig. 3 Components of strike velocity in meridional and front views (with dash lines the direction of a streamline): (a) meridional view of the diffuser and (b) front view of the diffuser

Measures To Reduce Strike Damage

Traditional pump designs exhibit diffusers with the leading edges of their vanes at right angles to the upcoming flow. If such a pump operates at a head above H_s, the absolute velocity at the diffuser inlet will exceed 4.8 m/s. To improve fish friendliness, the strike probability and mutilation ratio are the two factors that are to be reduced [18–20].

A predictive model presented by Kraker's [22] shows that strike probability is proportional to the number of diffuser vanes. Fish-friendliness can thus be improved by using a diffuser with fewer vanes. What is important in this respect, is that the solidity σ of a diffusing channel (defined as the ratio of vane length to pitch) should be sufficiently large to guide the flow and recover pressure [30]. It means that the streamwise length and the vane number of the diffuser should match.

As for the mutilation ratio, the strike velocity can be reduced considerably by not placing the leading edge at right-angles to the streamlines, but at a shallow angle instead. Fish will then be redirected to slide along the leading edge without a fatal strike. A practical utilization is the screw centrifugal pump with a spiral-shaped blade leading edge used to transport live fish with fairly low damage rates. The strike velocity v_s in Eq. (1) is defined as the velocity component perpendicular to the leading edge of the vanes [22]. As shown in Fig. 3, suppose that the diffuser vane leading edge is oriented at an angle α to the streamlines in the front view and at an angle of β in the meridional view. Following Eq. (2), the strike velocity is then given by

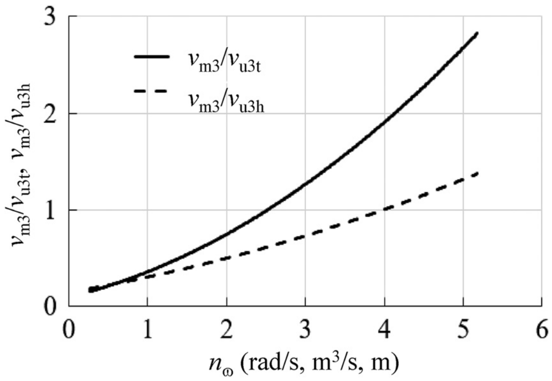

Fig. 4 The ratio of v_{m3} to v_{u3} at hub and tip of the diffuser, as a function of specific speed n_ω (defined as $\omega Q^{0.5}/(g{\cdot}H)^{0.75}$)

$$v_s = \sqrt{v_{u3}^2 \sin^2\alpha + v_{m3}^2 \sin^2\beta} \quad 0 < \alpha < 180\,\text{deg} \quad \text{and} \quad 0 < \beta < 180\,\text{deg} \tag{12}$$

The shape of the leading edge affects the strike velocity in a significant way. Which of the two angles, α or β, best to adapt to reduce the strike velocity, depends on the specific speed of the pump. According to Eqs. (3)–(5), the ratio v_{m3}/v_{u3} ranges between the values at the hub and the tip, given by Eqs. (13) and (14)

$$\frac{v_{m3}}{v_{u3h}} = \frac{2K_{m2}K_u h_2}{1 + s\sin\varepsilon} \tag{13}$$

$$\frac{v_{m3}}{v_{u3t}} = \frac{2K_{m2}K_u}{1 + s\sin\varepsilon} \tag{14}$$

where the subscript h and t for v_{u3} denote hub and tip of the diffuser, respectively.

In Fig. 4, the velocity ratios increase with the specific speed at both the diffuser hub and tip, indicating that the strike velocity is mainly in the circumferential direction for a low specific speed pump. Therefore, the strike angle α plays a dominant role in lowering the strike velocity outweighing the strike angle β. For high specific speed pumps, however, the focus is shifted to angle β for improving fish friendliness. In addition, an expanded spacing s between impeller and diffuser leads to velocity drop, unless $\varepsilon = 0$ is purely axial flow pumps. To pump manufacturers, however, an excessive gap produces unnecessary friction losses and the increased size of the pump also means more expensive fabricating costs. These factors are taken into account and balanced in the following design process.

Fish-Friendly Design With Vaned Diffuser

Prototype Pump. An existing mixed-flow impeller is studied as an example, as shown in Fig. 5. It has a single-bladed impeller with a spiral-shaped leading edge, designed to minimize damage to fish. It is combined with a concrete volute and operates at a specific speed n_ω of 1.85 (rad/s, m^3/s, m) at the best efficiency point. This pump is manufactured in different sizes with its inlet diameter ranging from 300 mm to 1250 mm, for heads up to 9 m. From a fish-friendliness point of view, such a volute normally leads to low damage to fish, in contrast to a multivane diffuser [4]. A drawback, however, is its radial dimension, as it is more than four times the size of the impeller inlet. This design is relatively expensive and space-consuming for installation, especially when the pump is increased in size to provide higher flow rates. One has to realize that an alternative pump of lower specific speed may provide equal or better fish handling capabilities [21], but its size will be even larger.

(*a*)

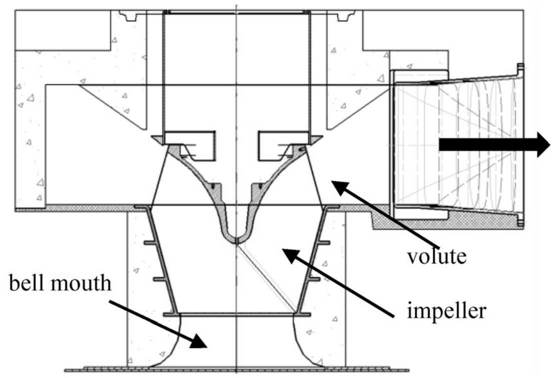

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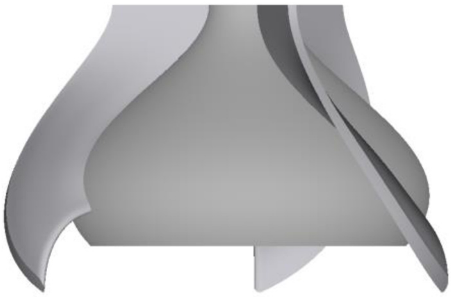
Fig. 5 Illustration of the single-bladed, mixed-flow impeller with a volute: (*a*) spiral-shaped impeller and (*b*) sectional view of the pump

Fish-Friendly Vaned Diffuser. To make the whole system more compact, a fish-friendly vaned diffuser was designed to replace the volute for the single-bladed impeller. The main design parameters and features of the vaned diffuser are as follows.

(1) The absolute velocities entering and leaving the diffuser are labeled as v_3 and v_4, respectively. To avoid excessive deceleration resulting in flow separation and stall, the de Haller ratio defined as v_4/v_3 should be above a certain critical value. The diffuser is designed to have a gradual increase in section area of the blade passage from the diffuser entrance to the exit. This way, the absolute velocity reduces gradually while a minimum de Haller ratio of 0.72 is maintained [30].

(2) The hub and shroud are designed as a bowl to redirect the flow and to minimize the radial dimension of the diffuser. The distance between the impeller blades and diffuser vanes at midspan is nearly half the size of the mean impeller exit diameter (gap ratio s equals 0.4). The leading edge of the diffuser is placed near the maximum radius of the bowl as shown in Fig. 6 where the circumferential component of the strike velocity has its minimum value and the ratio of v_m to v_u increases from 0.28 at the hub of the impeller exit to 0.55 in a vaned diffuser.



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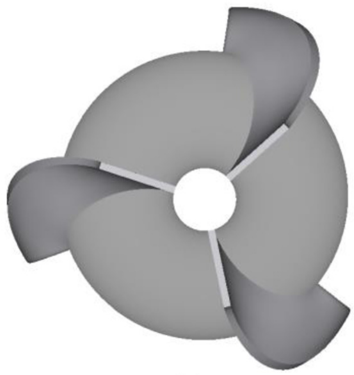


(*b*)

Fig. 6 Three-dimensional model of the fish-friendly vaned diffuser: (*a*) meridional view and (*b*) frontal view

(3) In conventional designs, seven or more vanes are normally recommended for the diffuser [28]. To reduce strike probability and improve fish friendliness [4], the number of vanes is set to three, and correspondingly, the diffusing channel is extended to achieve sufficient flow deflection. In this case, the solidity of diffuser vanes, σ (defined as the ratio of vane length to pitch) is in a reasonable range and between 1 and 1.3 [29], as shown in Fig. 7.

(4) The inlet vane angle, defined as the angle between circumferential direction and the tangent of the mean streamline on vane surfaces, matches the streamlines at 120% rated flow to enhance the discharge capacity for this particular pumping station. At the exit, the vane angle is designated to 90 deg.

(5) The vane leading edges are placed at a nonright angle to the streamlines near to the shroud, but perpendicular to the hub as demonstrated in Fig. 6. Such a curved leading edge, on the one hand, improves the uniformity of the gap between impeller blades and diffuser vanes. On the other hand, a reduction of meridional velocity is achieved near to the shroud and a sharp corner leading to fish being stuck in the wedge at the diffuser hub is also prevented.

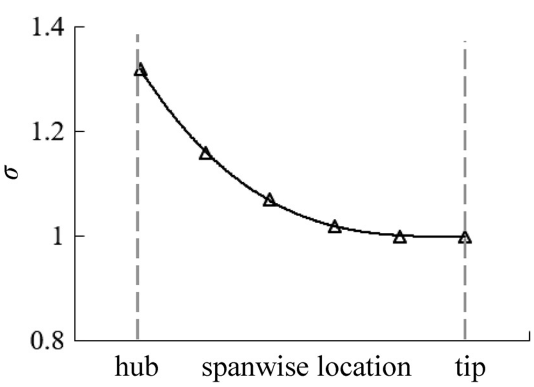
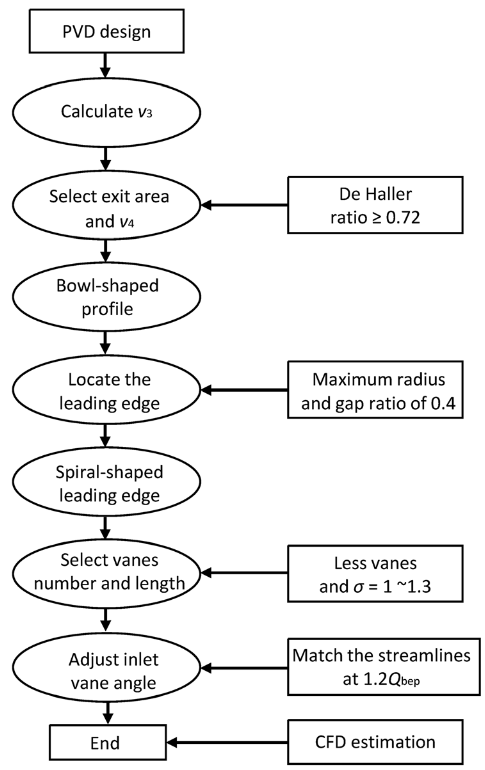


Fig. 7 Distribution of solidity in the spanwise direction

To better understand how the fish-friendly vaned diffuser is designed and the parameters are selected, the flowchart of the design procedure and the main parameters are exhibited in Fig. 8. Once the design of the vaned diffuser is completed, the CFD-based calculation will be carried out to check the performance and flow field, such that the design parameters can be optimized to achieve better performance.

Performance Comparison and Validation

Hydraulic Performance Test. To validate the hydraulic performance, the designed vaned diffuser is manufactured to match an impeller with an inlet diameter of 500 mm, and performance tests are carried out to compare it with the pump equipped with



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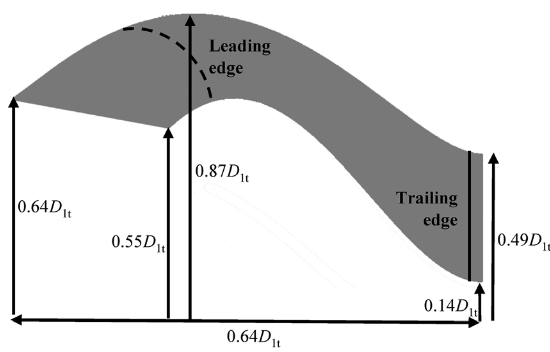

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Fig. 8 Design procedure and the main parameters of fish-friendly vaned diffuser: (*a*) design procedure and (*b*) main parameters, nondimensionalized with the impeller inlet tip diameter D_{1t}

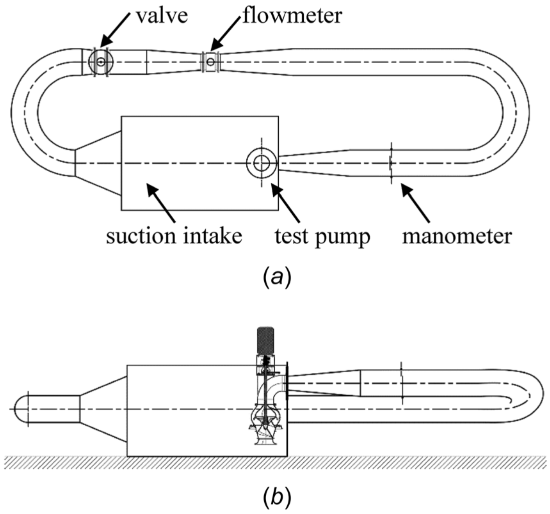

Fig. 9 Schematic arrangement of the test rig: (*a*) top view and (*b*) front view

the original volute. As shown in Fig. 9, the test bench is a loop system that consists of an open suction intake with a length of 6 m and a width of 4 m, where the pump is placed at one end. Before the test starts, the pump is placed and the tank is filled with water. The pump speed is controlled by a frequency converter and the flow is adjusted with the valve. When the flow and pressure have reached a stable value, the data are recorded. During the test, the volumetric flow rate and discharge pressure are measured by an electromagnetic flowmeter and a manometer, and the power transmitted to the impeller shaft is measured directly by a torque meter equipped with strain gages.

The uncertainty analysis of the experimental data is carried out according to the ISO9906-rotodynamic pumps-hydraulic performance acceptance tests. The overall uncertainty of measurement e consists of random uncertainty e_R and instrumental uncertainty (systematic uncertainty) e_S, given by

$$e = \sqrt{e_R^2 + e_S^2} \tag{15}$$

The estimation of random uncertainty e_R is calculated from the mean and standard deviation of the observations, written as

$$e_R = \frac{St}{\bar{x}\sqrt{n}} * 100\% \tag{16}$$

where S is the standard deviation, $\bar{x}$ is the arithmetic mean, n is the number of observations and five readings are recorded at the design flow rate, t is a function of n and equal to 2.78 based on a 95% confidence level.

The overall uncertainty of pump efficiency at design flow rate is then calculated using Eq. (17), and the step-by-step information can be found in Table 2. According to the ISO9906, the overall uncertainty of efficiency reaches the grade level 1

$$e_\eta = \sqrt{e_Q^2 + e_H^2 + e_p^2} \tag{17}$$

The performance curves of the head and efficiency of the pump with a vaned diffuser (PVD) and the pump with volute (PV) are presented in Fig. 10. The flow coefficient Q_c, head coefficient H_c and efficiency η_c are defined as

$$Q_c = \frac{Q}{ND_{1t}^3} \tag{18}$$

$$H_c = \frac{gH}{N^2 D_{1t}^2} \tag{19}$$

Table 2 Uncertainty analysis of the performance tests

	Q(m^3/h)	H(bar)	P(kW)
Test 1	2664.0	0.425	46.69
Test 2	2664.0	0.430	46.74
Test 3	2661.0	0.430	46.79
Test 4	2670.0	0.430	46.64
Test 5	2669.0	0.430	46.54
$\bar{x}$	2665.6	0.429	46.68
S	3.782	0.002	0.095
e_R (%)	±0.176	±0.648	±0.254
e_S (%)	±0.5	±0.6	±0.1
e (%)	±0.53	±0.88	±0.27
e_η (%)		±1.06	

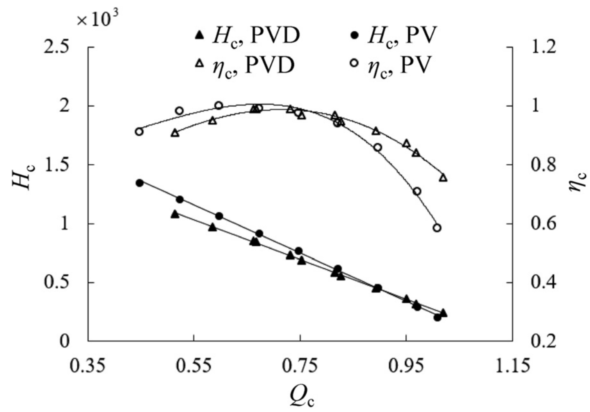

Fig. 10 Performance comparison between PV and PVD

$$\eta_c = \frac{\rho g Q H}{P \eta_{bep}} \tag{20}$$

where N and P are the shaft speed and power, Q is the volumetric flow rate, ρ is the density and η_{bep} is the peak efficiency of PV.

Considering the efficiency, the peak efficiencies of PV and PVD are at more or less the same level, and an increment of BEP flow rate of roughly 20% is found for the PVD. As described above, it is attributed to the diffuser inlet vane angle which was designed to accommodate this increase of the discharge capacity for the intended pumping station. Compared with the flow angle at rated flow, a steeper inlet vane angle will shift the BEP to a higher flow rate since the secondary flow loss increases at a larger incidence angle [31]. Accordingly, the pump efficiency and head of PVD slightly drop at lower flow rates and increase at higher flow rates, as can be observed in Fig. 10. In practical applications, such a deviation of performance is not an issue because the given pump can be increased in size or run at a higher shaft speed to provide the required head.

Fish Friendliness Validation. The experimental flow rate and head of the PVD at BEP are used to calculate the strike velocity at the leading edge of the diffuser vanes according to Eq. (12), in which the meridional and circumferential components are estimated with Eqs. (3) and (4). The striking velocity and its components are normalized with the pump head and presented in Fig. 11. For geometrically similar pumps operating at kinematically similar conditions (i.e., at similar operating conditions), the impeller tip speed u_{2t} and meridional velocity v_{m2} only depend on the required pump head. Internal flow angles and strike angles α and β in Eq. (12) are also similar. As a result, the ratio of strike velocity v_s and $(gH)^{0.5}$ remains constant as well for similar pumps

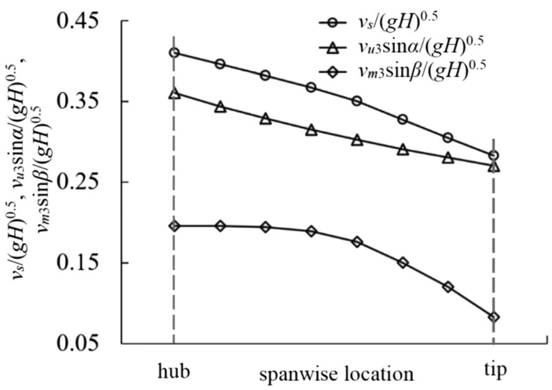

Fig. 11 Distribution of strike velocity and its components on the leading edge of the vaned diffuser at BEP, normalized with pump head

at similar operating conditions. Consequently, Fig. 11 is valid for any geometrically similar PVD pump operating at BEP, regardless of the size or the shaft speed. As seen in Fig. 11, the maximum dimensionless strike speed $v_s/(gH)^{0.5}$ in the diffuser is equal to 0.41. Suppose that the maximum allowed strike velocity is set to 4.8 m/s to attain a mutilation ratio of zero in the vaned diffuser, according to Eq. (1), the maximum head for the pump will be approximately 14 m. This improvement results from the combined effects of the one-bladed impeller that results in a larger impeller diameter and a lower v_{u2} compared with a conventional design, the shape of the vaned diffuser, and the shift of BEP to a slightly higher flow rate and lower head. As mentioned above, flood control pumping stations in the Netherlands normally have a lower head than 14 m. It means that the redesigned pumps will not show additional fish damage in the vaned diffuser.

Two years ago, a fish-friendly PVD with an inlet diameter of 50 cm was put into service in pumping station Obdam in The Netherlands (Fig. 12). Tests with live fish were carried out in this pumping station in which fish injury and mortality ratios of both cyprinids and eel were measured. Results of these tests are given in Table 3. In this paper, the results of the Obdam trials are compared with previously conducted fish tests for a PV with an inlet diameter of 30 cm (Ref. [12]) of which the results are summarized in Table 4. The pumps in both trials, PV and PVD, have geometrically similar impellers but different shaft speeds and diameters. The comparison between the two is based on the scaling law of fish damage by impeller blade strike which states that fish damage rates are equal for geometrically similar pumps if their values of L_f/D, Q/D^2, and H are the same [12]. Noting that, to eliminate the influence of the test setup as much as possible, the control groups (51 cyprinids and 50 eels for the PV, and 134 cyprinids and 86 eels for PVD) were set to force fish to undergo a similar treatment without passing through the pumping system, including transportation to the test site, netting out of the recovery tank, and retention in an aerated holding tank [12], and the recorded fish mortality were deducted from the test results. Besides, both tests were done according to the NEN 8775:2020 standard [32].

Fig. 12 Fish friendly vaned diffuser in Obdam pumping station in The Netherlands

Table 3 Fish test of PVD, with P_m the mortality ratio and n_f the number of fish

Test condition	1	2	3
N (r/min)	400	438	483
Q (m^3/min)	37	43	51
H (m)	3.8	4.1	4.5
Q/D^2 (m/min)	135	157	186
n_f, cyprinids	60	107	103
L_f/D, cyprinids	0.44	0.45	0.46
P_m (%), cyprinids	5	4.7	9.7
n_f, eel	105	100	100
L_f/D, eel	1.55	1.53	1.53
P_m (%), Eel	0	2	0

Table 4 Fish test of PV, with P_m the mortality ratio and n_f the number of fish [12]

Test condition	1	2	3	4	5	6	7	8	9	10
N (r/min)	360	580	870	510	610	730	870	610	740	870
Q (m^3/min)	6.9	13.5	21.5	7.2	10.8	14.8	19.1	8.1	12.9	17.3
H (m)	0.96	1.49	2.53	2.84	3.15	3.54	4.12	4.28	4.72	5.26
Q/D^2 (m/min)	77	150	239	80	120	164	212	90	143	192
n_f, cyprinids	46	74	107	62	44	59	70	55	67	57
L_f/D, cyprinids	0.45	0.47	0.46	0.48	0.48	0.48	0.46	0.48	0.47	0.48
P_m (%), cyprinids	2	4	26	2	9	10	34	42	12	37
n_f, eel	46	56	51	49	52	50	52	46	58	51
L_f/D, eel	1.55	1.56	1.52	1.57	1.59	1.58	1.60	1.60	1.62	1.60
P_m (%), eel	2	2	14	0	0	0	2	0	2	6

Tables 3 and 4 show that the ratio of mean fish length and impeller diameter L_f/D are quite close for the PV and PVD tests: 0.47, respectively, 0.45 for cyprinids and 1.58, respectively, 1.54 for eel. Thus, fish damage rates for both pumps can be illustrated in a graph of H versus Q/D^2 in Fig. 13, where the mortality rates are indicated by the areas of the circles. Overall, for both cyprinids and eel, the fish damage caused by the PVD is smaller than that of the PV. It shows that fish friendliness seems to have improved for a PVD due to the replacement of a traditional volute by a so-called fish-friendly vaned diffuser. In addition, the good hydraulic performance and its smaller size make the PVD more competitive in practical use.

Conclusions

This study presents an analysis of the strike damage to fish by diffuser vanes of a pump. The correlation between the strike velocity at the leading edge of the vane and the pump head was presented as a function of specific speed. It shows that the strike velocity has its maximum near to the hub and increases with the head for a certain specific speed of the pump. Besides, pumps

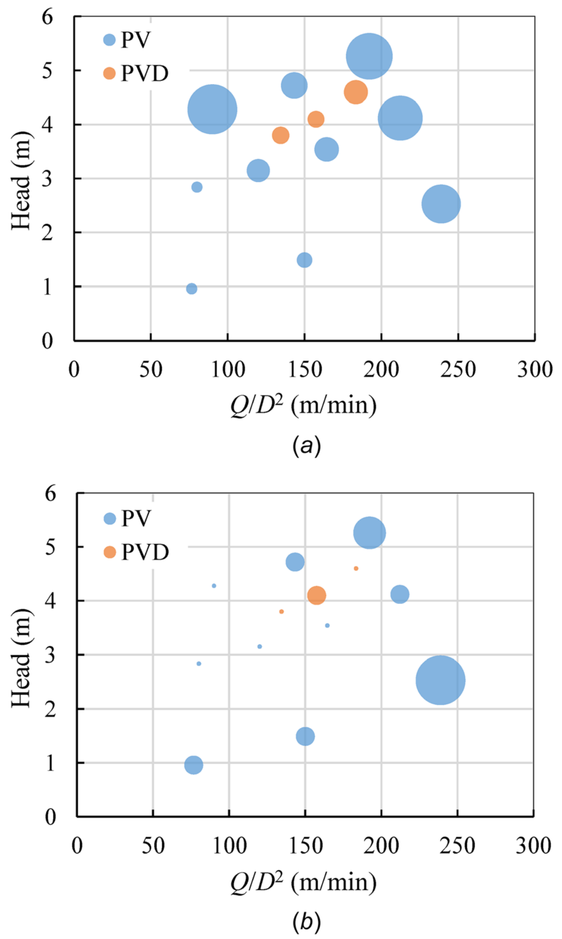

Fig. 13 Comparison of fish damage between PV and PVD, where the area of the circles denotes the fish mortality rates. Data are taken from Tables 3 and 4, and the smallest circles in Fig. 12(*b*) denote mortality of 0%: (*a*) mortality of Cyprinids with length $L_f \sim 0.46D$ and (*b*) mortality of eel with length $L_f \sim 1.57D$.

with lower specific speed can operate to a higher head without fish strike mutilation.

One way to reduce the strike damage in the vaned diffuser is to reduce the strike probability by using fewer vanes and extending the flow channels such that the solidity is sufficiently large to achieve the required flow deflection. Another way is to lower the strike mutilation ratio by not placing the leading edge at right angles to the streamlines.

For a low specific speed pump, the strike velocity onto the diffuser vanes mainly results from the circumferential component and thus the focus for reducing strike velocity is on restraining the projected shape of the leading edge of the vane in the front view plane. For a high specific speed pump, the strike angle in the meridional view plays a more important role. Additionally, increasing the gap between impeller blades and diffuser vanes causes strike velocity to reduce for radial and mixed flow pumps.

A vaned diffuser was designed to operate with a mixed-flow impeller, with the aim of obtaining a high degree of fish friendliness. It was found that, if a zero fish mutilation ratio is required in the diffuser, the diameter and the shaft speed of this pump can be selected to operate at maximum efficiency with heads of up to 14 m, which is sufficient for practical applications of this flood-relief pump. Such an improvement of fish friendliness can be attributed to several points:

(1) the diffuser vane leading edge is placed near the maximum radius of the bowl to minimize the circumferential component of the strike velocity.
(2) the spiral-shaped leading edge is placed at an acute angle to both streamlines and circumferences to reduce the strike velocity component perpendicular to the leading edge.
(3) the BEP is shifted to a slightly higher flow rate, leading to the drop of the pump head as well as the strike velocity component v_{u2}.
(4) the expanded spacing between rotating impeller and stationary diffuser may result in less damage to eels caused by the scissor effect. The duration for eels lingering in the hazardous region is reduced with an expanded gap, and the extreme case is for cyprinids that the streamwise-projected fish length is shorter than the gap size which will not cause a shear effect.

The performance of the fish-friendly design with vaned diffuser was measured and compared with the original design based on the same impeller but equipped with a volute. The comparison included both hydraulic performance and fish handling performance. It showed that the peak efficiencies of the original pump and the pump with the vaned diffuser were at more or less the same level, while an improvement of fish friendliness was achieved by using a so-called fish-friendly vaned diffuser.

Funding Data

- China Scholarship Council (File No. 201608320265; Funder ID: 10.13039/501100004543).
- National Natural Science Foundation of China (Grant Nos. 51979125 and 51979138; Funder ID: 10.13039/501100001809).
- Water Conservancy Science and Technology Project of Jiangsu Province (Grant No. 2021007; Funder ID: 10.13039/501100018581).
- Jiangsu Provincial Science Fund for Distinguished Young Scholars (Grant No. BK20211547).
- Excellent Scientific and Technological Innovation Team Project in Colleges and Universities of Jiangsu Province (Grant No. SKJ(2021)-1; Funder ID: 10.13039/501100013085).

Nomenclature

$D_{1\mathrm{h}}$ = impeller inlet hub diameter (m)
$D_{1\mathrm{t}}$ = impeller inlet tip diameter (m)
D_2 = impeller exit radius (m)
$D_{2\mathrm{h}}$ = impeller inlet hub diameter (m)
$D_{2\mathrm{m}}$ = geometric average of diameters at impeller exit (m)
$D_{2\mathrm{t}}$ = impeller inlet tip diameter (m)
d = leading edge thickness (m)
e = overall uncertainty of measurement
e_{R} = random uncertainty
e_S = instrumental uncertainty (systematic uncertainty)
e_η = overall uncertainty of efficiency
g = gravitational constant (m/s^2)
H = theoretical head (m)
h_2 = hub ratio
H_c = head coefficient
H_s = theoretical head with no strike injury to fish in the diffuser (m)
K_d = maximum strike velocity coefficient at diffuser inlet
K_u = tip velocity coefficient at impeller exit
$K_{\mathrm{m}2}$ = meridional velocity coefficient at impeller exit
L_g = gap length (m)
L_f = fish length (m)
n = number of observations
n_f = number of fish
n_ω = specific speed, defined as ωQ0.5/(gH)0.75
N = shaft speed (r/min)
P = shaft power (W)
P_m = mortality ratio (%)
Q = volumetric flow rate (m^3/s)
Q_c = flow coefficient
s = gap ratio

S = standard deviation
u_{2t} = tip velocity at impeller exit (m/s)
v_s = strike velocity (m/s)
v_{m2} = meridional component of absolute velocity at impeller exit (m/s)
v_{m3} = meridional component of absolute velocity at diffuser entrance (m/s)
v_{u2} = circumferential component of absolute velocity at impeller exit (m/s)
v_{u3} = circumferential component of absolute velocity at diffuser entrance (m/s)
$v_{u2\,h}$ = circumferential component of absolute velocity at hub of impeller exit (m/s)
v_3 = absolute velocity at diffuser entrance (m/s)
$v_{3\,h}$ = absolute velocity at hub of diffuser entrance (m/s)
α = impact angle in impeller frontal view (rad)
β = impact angle in impeller meridional view (rad)
ε = angle between outflow and rotating axis in the cross section of impeller (rad)
η_{bep} = peak efficiency of prototype pump (%)
η_c = efficiency coefficient
ρ = density (kg/m^3)
ω = angular velocity (rad/s)

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