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PECULIARITIES OF CALCULATION OF THE MAIN PARAMETERS OF THE WATER JET PROPULSION SYSTEM

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A set of basic hydrodynamic parameters of the water jet propulsor is proposed. An analytical method for determining the optimal values of design parameters, at which the maximum efficiency of the propulsor is achieved, is given, as well as a method for calculating parameters other than hydrodynamically optimal, which expands the developer's possibilities for optimizing the dimensions and mass of the propulsor. The key hydrodynamic parameters for water jet propulsor design are highlighted. The relationship between the hydrodynamic parameters of the water jet propulsor is shown. The meaning of the main hydrodynamic parameters is explained. The physical meaning of optimal values of hydrodynamic parameters is shown. The presented method of determining the limiting values of parameters at the design point is based on experimental studies, which makes it possible to establish the boundary of the area of existence of parameters of such propulsors. This method, together with the determination of optimal parameters by the value of the coefficient of efficiency and the same coefficient at suboptimal parameters, completes the problem of selecting the water jet propulsion's design parameters. The method allows solving such important problems as creating a propulsor that develops maximum pressure at a given circular velocity, or a propulsor with minimum diameter but maximum static efficiency, etc. The methodology is based on the method of determining optimal and limiting design parameters using the Bernoulli equation. Definitions of basic hydrodynamic characteristics are given. The method of calculation by the method is shown on the example of the simplest case of the mover with one fan, when the diameters of the inlet and outlet channels are the same. The result of the calculations is illustrated graphically. The method of comparing design and experimental data of propulsors having different hydrodynamic schemes with equal and different design parameters allows for the revelation of the peculiarities of their characteristics and to apply in each specific case one or another scheme of the propulsor fan. The proposed method allows for the development of a propulsor for a watercraft under the given requirements for it, as well as to select a ready-made propulsor from the existing catalogs for a specific task of a watercraft. The features of calculating propulsors in the form of tubular axial fans with propellers in a ring are considered more efficient than propellers, allowing a gain in dimensions of a swimming vehicle. Besides, it is safer and more reliable in operation than a propeller.

Key words: *Propeller, efficiency factor, parameter, limit values, design point, pressure, diameter, Bernoulli equation, fan, hydrodynamic scheme, ring, dimensions, reliability, efficiency.*

Сєлюков О., Хаолін Ліу. Особливості розрахунку основних параметрів водометного рушія

Пропонується комплекс основних гідродинамічних параметрів водометного рушія. Наведено аналітичний метод визначення оптимальних значень розрахункових параметрів, за яких досягається максимальний коефіцієнт корисної дії рушія, а також метод розрахунку параметрів, відмінних від гідродинамічно оптимальних, що розширює можливості розробника щодо оптимізації габаритів і маси рушія. Виділено ключові гідродинамічні параметри для проектування водометного рушія. Показано зв'язок між гідродинамічними параметрами водометного рушія. Роз'яснено значення основних гідродинамічних *параметрів. Показано фізичний сенс оптимальних значень гідродинамічних параметрів. Представлений метод визначення граничних значень параметрів у розрахунковій точці ґрунтується на експериментальних дослідженнях, що дає змогу встановити межу області існування параметрів таких рушіїв. Цей метод разом із визначенням оптимальних параметрів за величиною коефіцієнта корисної дії, а також цього ж коефіцієнта за неоптимальних параметрів завершує завдання вибору розрахункових параметрів водометного рушія. Метод дає змогу розв'язувати такі важливі завдання, як створення рушія, що розвиває максимальний тиск за цієї кругової швидкості, або рушія з мінімальним діаметром, але з максимальним статичним коефіцієнтом корисної дії тощо. В основу методології покладено метод визначення оптимальних і граничних розрахункових параметрів з використанням рівняння Бернуллі. Дано визначення основних гідродинамічних характеристик. Методику розрахунку за методом показано на прикладі найпростішого випадку рушія з одним вентилятором, коли діаметри вхідного і вихідного каналів однакові. Результат розрахунків проілюстровано графічно. Метод зіставлення розрахункових і експериментальних даних рушіїв, що мають за рівних і різних розрахункових параметрів різні гідродинамічні схеми, дає змогу виявити особливості їхніх характеристик і застосовувати в кожному конкретному випадку ту чи іншу схему вентилятора рушія. Запропонований метод дає змогу розробляти рушій для плавального засобу під задані вимоги до нього, а також підібрати готовий рушій з наявних каталогів під конкретне завдання плавального засобу. Розглянуто особливості розрахунку рушіїв у вигляді трубчастих осьових вентиляторів із пропелерами в кільці, які є ефективнішими, ніж гребні гвинти, що дає змогу отримати виграш у габаритах плавального засобу, до того ж безпечніший і надійніший у процесі експлуатації, ніж гребний гвинт.*

Ключові слова: *Рушій, коефіцієнт корисної дії, параметр, граничні значення, розрахункова точка, тиск, діаметр, рівняння Бернуллі, вентилятор, гідродинамічна схема, кільце, габарити, надійність, ефективність.*

Introduction. Water jet propulsion for river and marine types of watercraft is widely used worldwide [1]. This propulsion system is used [2] in the form of high-power multi-shaft units on giant high-speed ferries or on the smallest high-speed vehicles like jet skis, but in both cases, the speed parameter is an advantage. Waterjets have high friction costs as the water moves through the propulsor, but this disadvantage is compensated for by the increased efficiency of the impeller. As a result, in terms of its propulsive characteristics, the water jet is practically equal to the propeller, and at high navigation speeds, it has an advantage [3]. Another advantage of the waterjet is the amazing softness of the transmission operation and almost complete absence of vibration [4], which is not unimportant for recreational watercraft. Waterjet propulsors in the market occupy no more than 10 % of the total market volume of propulsion devices [5]. The hydrodynamic calculation of the propulsion system plays a significant role in the design of a swimming vehicle [6]. Axial fans in the form of propellers in a ring are usually used as water jet propulsors. Axial fans work by drawing water into the fan and moving it in a parallel direction along the axis of the blades. The simplicity of their design and efficiency make them a popular means of creating propulsion for watercraft. Axial fans consist of one or more rows of blades (impellers) attached to a central hub (hub). An intermediate guide apparatus may be placed between the rows of blades. An inlet guide apparatus may be located in front of the first impeller. Behind the last impeller, there may be a straightening device. The intermediate, inlet guide and straightening apparatuses are designed to change the direction of the water. There are three basic types of axial fan design: propeller fans, tubular axial fans, and vane axial fans. Propeller fans, such as propellers, are simple low-pressure axial fans that move large volumes of water at low pressure. Tubular axial fans (channel propulsors) are propeller fans that are enclosed in a cylindrical housing (channel) better to direct the water flow (water cannons). Bladed axial fans are equipped with guide vanes at the fan outlet to improve efficiency and control the direction of water flow, and are designed for high pressure. The water-jet propulsor is more efficient than the propeller fan [7], which allows for a gain in its dimensions. Besides, it is safer and more reliable since the impellers are shielded from damage by a solid pipe. Increased efficiency of the propeller also allows such fans to be fitted with protective grilles at the front and rear of the propeller, which also contributes to increased reliability and safety in the operation of the propeller. The fan's operation principle is based on creating a pressure difference between its inlet and outlet. The fan blades rotate at speed, creating a low pressure at the inlet and a high pressure at the outlet. This pressure difference creates the movement of the water flow through the fan and generates thrust for the propulsion of the watercraft.

Formulation of the problem. The axis of the complex of the main hydrodynamic parameters is not defined for the water jet propulsion system and the relationship between them is not shown. A practical calculation method for determining the optimal values of the design parameters is not provided and their physical meaning is not shown, including those at which the maximum efficiency of the thruster is achieved. The key hydrodynamic parameters are not identified for the design stage of the waterjet propulsion system.

The aim of the study. To provide an analytical method for determining the optimal values of design parameters that achieve the maximum efficiency of the thruster and a method for calculating parameters other than hydrodynamically optimal, which expands the developer's ability to optimize the dimensions and weight of the thruster. To highlight the key hydrodynamic parameters for the design of a water jet propulsion system. Show the relationship between the hydrodynamic parameters of a water jet propulsion system. Explain the meaning of the main hydrodynamic parameters. Show the physical meaning of the optimal values of hydrodynamic parameters.

Analysis of recent research and publications. In article [2] a general analysis of the efficiency of water jet propulsion on watercraft without reference to hydrodynamic parameters is given. In [3] the efficiency of water jet propulsion is investigated by modeling without analytical methods. In [4], only noise from water jet propulsion is investigated without investigating other parameters. In [5], the world's water jet propulsion research methods are analyzed without disclosing the peculiarities of the analytical method. In [7], the hydrodynamic calculation of only an open propeller is given.

Presentation of the main research material. Usually, the hydrodynamic calculation [8] of axial fans consists of the following main steps:

– selection of the design scheme [9] and determination of the design parameters of the fan, including diameter and speed, based on the given values of pressure, performance and pressure losses in the elements that make up the propulsor;

– calculation of flow, flow kinematics, velocity triangles in front of the blade crowns and behind them along the radius. Determination of the geometry of the blade crowns (profiling) allows to realize a given flow at the design point at the lowest pressure loss, i.e. with the highest efficiency;

– calculation of hydrodynamic characteristics of the fan in the operating range of its capacity;

– calculation of hydrodynamic characteristics of the fan at its regulation.

One of the most important metrics on which the developers of swimming vehicles are oriented is their hydrodynamic characteristics. The main parameters of hydrodynamics are such indicators as: water flow, thrust, performance, static and dynamic pressure, energy consumption, and power. In general, the characteristic of a water fan is the dependence of the total pressure, power on the shaft and coefficient of efficiency on the water flow rate at a constant speed of rotation of the impeller of known size and known water density and hydrodynamic scheme, i.e. the totality of the geometric configuration of the flow part and impeller. Hydrodynamic characteristics of the propulsor are necessary both at the stage of design and at the stage of its testing. Hydrodynamic characteristics are distinguished between absolute (pressure, performance, power, etc.) and relative in dimensionless parameters, when, for example, instead of pressure, the pressure coefficient is used, instead of performance, the performance coefficient, and instead of power, the power coefficient.

Due to the variety of requirements that designers have for watercraft, it is often necessary to develop a hydrodynamic scheme to ensure that the fan is manufactured to best suit its layout and application.

The relationship between the hydrodynamic parameters can be seen if the expression for the thrust F of the fan propulsor using the equation of quantity of motion is presented in this form:

$$F = \int_S \rho v_{out}(v_{out} - v)dS, \qquad (1)$$

where w_{out} – speed at the mover outlet, m/s;
v – speed of the swimming vehicle, m/s;
ρ – water density, kg/m^3;
S – channel area, m^2.

It is more convenient to consider the value of these parameters in Fig. 1 for the simplest case of a propulsor with one fan, when the diameter of the inlet channel and the outlet channel are the same and the pressure outside the outlet channel is equal to the pressure in front of the inlet channel.

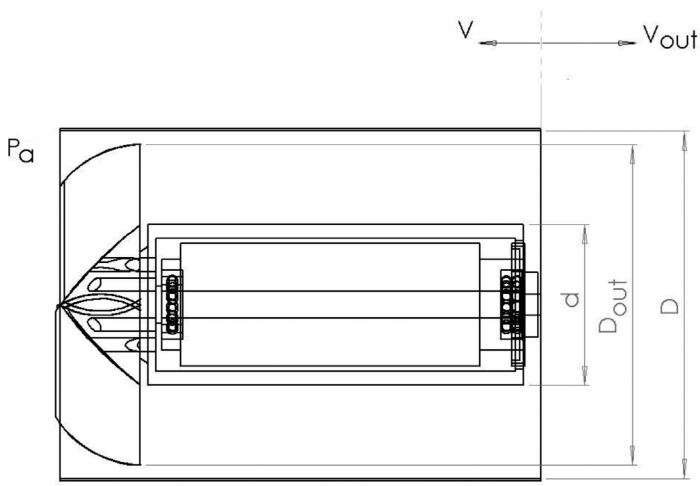

Fig. 1. Channel fan in section: p_a – pressure outside the mover; D – channel diameter, D_{out} – fan diameter ($D_{out} < D$), d – sleeve diameter, $S = \dfrac{\pi D^2}{4}$, $u = \dfrac{d}{D}$, $S_{out} = \dfrac{\pi D_{out}^2}{4}$

An expression for the thrust F of the fan propulsor through the parameters in the outlet channel cross-section using Bernoulli's equation:

$$F = \int_S \rho v_{out} v \left(\sqrt{\frac{2(p_{out} - p_a)}{\rho v^2} - \left(\frac{v_{out}}{v}\right)^2} - 1 \right) dS. \qquad (2)$$

Expression (2) makes it possible to determine the real thrust of the propulsor in the process of its testing. For practical application, this formula can be further simplified by neglecting the small value of the $\frac{v_{out}}{v}$ ratio, the change in velocity v_{out} along the cross-section S and the shift in density ρ in the propulsion path. In this case, the equation for thrust F to reveal the regularities of influence of the main parameters on it will be written in the following form:

$$F = \frac{\pi D^2}{4}(1-u^2)\rho C_{in}(v_{out}-v) = \rho Q(v_{out}-v), \quad (3)$$

where Q – water flow rate, m^3/s;

u – relative diameter of the impeller bushing;

C_{in} – is the axial component of water flow velocity at the propulsor inlet, m/s.

Flow velocity, defined as the volume of water moved per unit time, is a fundamental performance indicator for axial flow fans. The required flow rate depends on the specific application of the swimming apparatus.

In practice, in calculations, the thrust is specified not for one mode of operation of the watercraft, but for several, for cruising speed and maximum speed.

For fans, a distinction is made between static and dynamic pressure. Static pressure is related to the force of the head of water generated by the fan. Static pressure determines the back pressure the fan may encounter when pumping water through itself. This parameter is measured in pascals (Pa). The optimum static pressure value is selected based on the specific design of the watercraft and depends on the fan design and impeller speed. The higher the value, the better the fan performs under high drag conditions.

Dynamic pressure is the most critical parameter, as it affects the ability of the fan to overcome the resistance created by various obstacles when pumping water through itself. The higher the dynamic pressure value, the more resistance it can overcome and the more water it can move. It is measured in pascals (Pa).

Losses in the p_{can} fan duct are due to friction, leakage, and turbulence. Inefficiencies caused by water resistance, turbulence or improper blade design can increase energy consumption. Friction losses usually occur on guard grids, blade surfaces and other moving parts. They can be minimized by high workmanship and the use of low-friction materials. Water leakage around fan blades and housing can reduce effective flow and increase energy consumption. Improper installation of fan blades and the presence of a protective grille at the inlet of the propulsion channel can lead to turbulent water flow within the channel, which increases channel resistance and reduces propulsion efficiency.

The flow rate is the volume of water that passes through the fan per unit of time. The unit of measurement is cubic meter per second (m^3/s). The water flow rate depends on several factors such as fan blade speed, shape and number of blades, inlet and outlet pressure, and water temperature. At a given water flow rate Q, the fan should develop such a total pressure p_v that provides the water head with kinetic energy $E_k = \frac{\rho v_{out}^2}{2}$, overcoming the channel resistance at the channel inlet Δp_{in} and outlet Δp_{out}, as well as overcoming possible pressure losses Δp_{ins} associated with the placement of auxiliary elements of the channel structure: rudders, grids, deflectors, nozzles, etc., as well as overcoming possible pressure losses Δp_{ins} associated with the placement of auxiliary elements of the channel structure: rudders, grids, deflectors, nozzles, etc. In addition, changes in water density due to temperature fluctuations insignificantly affect the fan performance. Increasing the rotational speed or blade pitch may increase the flow rate, but these changes also affect other parameters such as power and efficiency. Considering these variables during the design phase is important to ensure consistent performance under different operating conditions.

Considering the optimal balance between pressure rise and flow velocity in the design is critical [10]. If the fan generates too little pressure, it will have difficulty moving water through the system, reducing efficiency. On the other hand, over-pressurization can lead to unnecessary energy consumption and noise. Choosing the correct blade design, fan speed, and operating conditions will ensure that the increase in pressure is sufficient to overcome the resistance in moving water through itself without compromising efficiency, while maintaining the desired flow rate.

The channel resistance in the general case is represented as follows:

$$p_v = \frac{\rho v_{out}^2}{2} + \Delta p_{in} + \Delta p_{out} + \Delta p_{ins}, \tag{4}$$

at that the channel resistance coefficients are a set:

$$\zeta = \{\zeta_2, \zeta_{in}, \zeta_{out}, \zeta_{ins}\}, \tag{5}$$

where $\zeta_2 = \dfrac{E_k}{\dfrac{\rho C_{in}^2}{2}}$; $\zeta_{in} = \dfrac{\Delta p_{in}}{\dfrac{\rho C_{in}^2}{2}}$; $\zeta_{out} = \dfrac{\Delta p_{out}}{\dfrac{\rho C_{in}^2}{2}}$; $\zeta_{ins} = \dfrac{\Delta p_{ins}}{\dfrac{\rho C_{in}^2}{2}}$.

Water flow exit area from the fan taking into account the relative diameter of the impeller sleeve:

$$S_{out} = \frac{\pi D^2}{4}(1-u^2). \tag{6}$$

Ratio of average velocities (areas) of water flow exit from the propulsor nozzle and from the fan:

$$n_S = \frac{S}{S_{out}} = \frac{C_{in}}{v_{out}} = \sqrt{\frac{1}{\zeta_2}} \tag{7}$$

and therefore the channel resistance in the general case

$$p_v = \left(\zeta_{sum} + \frac{1}{n_S^{\,2}}\right)\frac{\rho C_{out}^2}{2} = (1+\zeta_{sum}\, n_S^2)\frac{\rho v_{out}^2}{2}, \tag{8}$$

where $\zeta_{sum} = \zeta_{in} + \zeta_{out} + \zeta_{ins}$.

When calculating fans for swimming vehicles, considering their typical design, the influence of the compressibility of the water flow can be neglected.

The dynamic flow pressure on the fan associated with the motion of the watercraft is taken into account by means of the coefficient of utilization of this pressure α, where $\alpha < 1$, the value of which depends mainly on the layout and type of water intake, the velocity ratio $\dfrac{C_{in}}{v}$ and on the location of the propulsor in the channel. When the watercraft moves at velocity v, the dynamic pressure $\dfrac{\rho v^2}{2}$ or its part $\dfrac{\alpha\rho v^2}{2}$ also participates in overcoming the channel resistance, as does the total fan pressure p_v. When $\dfrac{C_{in}}{v} \geq 1$, the coefficient α is in the range $\alpha = 0{,}85\ldots0{,}95$.

Taking into account the pressure utilization factor α, the required fan pressure p_v is as follows:

$$p_v = \left(\zeta_{sum} + \frac{1}{n_S{}^2}\right)\frac{\rho C_{out}^2}{2} - \frac{\alpha \rho v^2}{2}. \tag{9}$$

The power requirements for axial flow fan design are key to the design process. Power consumption has a direct impact on both operating costs and the overall efficiency of the watercraft. Understanding the power requirements of an axial fan involves a clear understanding of the forces acting during its operation. The power required by an axial fan is directly proportional to flow rate and pressure rise and inversely proportional to efficiency. Higher flow rates and pressure rise increase the power requirements, while higher efficiency reduces the power required. Power N consumed by the fan:

$$N = \frac{Qp_v}{\eta}, \tag{10}$$

where η – fan efficiency.

Traction F through velocity v_{out}:

$$F = \rho Q(v_{out} - v). \tag{11}$$

Knowledge of the density of water is an essential factor in engineering design. It is used in various areas of engineering where the effects associated with the movement of liquids and gases need to be considered. Understanding water density is particularly important in the design and operation of watercraft. Water density varies at different depths, affecting swimming vehicles' water resistance. The density of water at various depths depends on several factors, including pressure, temperature, salinity, and degree of purity. According to the above formulas, the hydrodynamic parameters of the watercraft are directly related to the water density ρ. The thrust characteristic in calculations can be used in absolute values (H) or as dimensionless relative values, such as specific thrust (kg of thrust per one kg of water) or thrust factor $\bar{F}$.

Traction coefficient $\bar{F}$ through the speed of the watercraft:

$$\bar{F} = \frac{F}{\frac{\rho v^2}{2}\frac{\pi D^2}{4}(1-u^2)}. \tag{12}$$

Flow velocity at the fan outlet v_{out} through the draft factor:

$$v_{out} = \frac{v}{2}\left(1 + \sqrt{1 + \frac{2\bar{F}}{n_S}}\right). \tag{13}$$

Substituting v_{out} from (13) into (8), the total fan pressure through the thrust coefficient:

$$p_v = \frac{\rho v_{out}^2}{2}\left[\frac{1+\zeta_{sum}\, n_S^2}{4}\left(1 + \sqrt{1 + \frac{2\bar{F}}{n_S}}\right)^2 - \alpha\right]. \tag{14}$$

Through the traction coefficient, the performance formula is as follows:

$$Q = \frac{F\, n_S}{\bar{F}\rho v}\left(1 + \sqrt{1 + \frac{2\bar{F}}{n_S}}\right). \tag{15}$$

Efficiency is one of the most critical parameters in axial flow fan design because it directly affects energy consumption and operating costs. Efficiency indicates how efficiently the fan converts input power into practical work in moving water. The overall efficiency of an axial flow fan is affected by several factors, which are categorized into either the mechanical efficiency group or the hydrodynamic efficiency group. Mechanical efficiency refers to the efficiency of the mechanical components of the fan, such as the motor and bearings. Hydrodynamic efficiency refers to how efficiently the fan blades convert rotational energy into water flow. The design of the blade is crucial here, with factors such as angle of attack and blade shape playing an important role.

The energy consumption of a fan is directly related to the efficiency of its operation. It may depend on various factors, including: blade design, material of manufacture, technical parameters of the fan, and others. The optimal combination of energy consumption and fan performance is one of the main challenges in watercraft design. Energy consumption is affected by three factors:

– blade shape and size. The design of the fan blades, including their pitch, curvature and length, has a significant effect on the power required. Larger blades or blades with steeper angles require more power to move the same volume of water;

– motor characteristics. The type of motor selected also plays a role. High efficiency motors can reduce power consumption, while underpowered motors may not achieve the required performance.

The generalized external propulsor efficiency η_{outs} is designed to account for losses in the channel and the extent to which the kinetic energy of motion is utilized. Its physical meaning is the extent to which η_{outs} reflects the ratio of the useful power associated with the motion of the watercraft to the hydraulic power of the flow through the propulsor:

$$\eta_{outs} = \frac{\sqrt{1+\frac{2\bar{F}}{n_S}}-1}{\frac{1+\zeta_{sum} n_S^2}{4}\left(1+\sqrt{1+\frac{2\bar{F}}{n_S}}\right)^2-\alpha}. \tag{16}$$

The power through η_{outs} is expressed as follows:

$$N = \frac{Fv}{\eta\eta_{outs}}. \tag{17}$$

The method of determining the parameters of the propeller when only the fan diameter is known does not allow to judge the minimum possible power required to drive the fan. In contrast to the propeller, the built-in fan-driver has a maximum η_{outs} due to pressure losses associated with its location in the channel. This leads to the necessity to find the optimal parameters of the realizing fan corresponding to the minimum possible power of its drive.

As can be seen from the above formulas, the propulsion fan is calculated for one optimal speed for some considerations. If the speed changes, if the resistance coefficient of the channel or the rotation speed of the propulsion fan changes, the thrust force, power consumption and other parameters will change. Naturally, these changes must be taken into account when selecting or fabricating the right fan. From the equation $\partial\eta_{outs}/\partial\bar{F}$ for a given n_S, the optimum diameter D_{opt} of the propulsion fan can be found:

$$\overline{F}_{opt} = 2\sqrt{1-\frac{\alpha}{1+\zeta_{sum}\, n_S^2}}\left(1+\sqrt{1-\frac{\alpha}{1+\zeta_{sum}\, n_S^2}}\right) n_S. \tag{18}$$

$$D_{opt} = \frac{4F}{\pi\rho(1-u^2)\, n_S v^2 \left(1+\sqrt{1-\frac{\alpha}{1+\zeta_{sum}\, n_S^2}}\right)\sqrt{1-\frac{\alpha}{1+\zeta_{sum}\, n_S^2}}}. \tag{19}$$

The optimum water flow rate v_{out_opt} corresponding to the optimum diameter D_{opt} given v and n_S:

$$v_{out\_opt} = v\left(1+\sqrt{1-\frac{\alpha}{1+\zeta_{sum}\, n_S^2}}\right). \tag{20}$$

The physical meaning of the optimal value of D_{opt} at a given n_S is as follows. Decreasing D at a given thrust leads to the necessity of increasing the water flow velocity behind the fan. In this case, due to the increase in losses, the required p_v increases and Q decreases, since the increase in the water flow velocity behind the fan is less than the decrease in the area of the flow path. When D increases, the opposite is true. The minimum of the product $p_v Q$ corresponds to D_{opt}, which increases with decreasing n_S. This leads to decreasing flow velocities, decreasing pressure losses and increasing maximum η_{outs}. Fig. 2 shows the graph of $F_{opt}(\zeta)$ dependence at different n_S with lines of equal values of $\left(\frac{v_{out}}{v}\right)_{opt}$ corresponding to F_{opt}, where $\left(\frac{v_{out}}{v}\right)_{opt} = 1{,}85$.

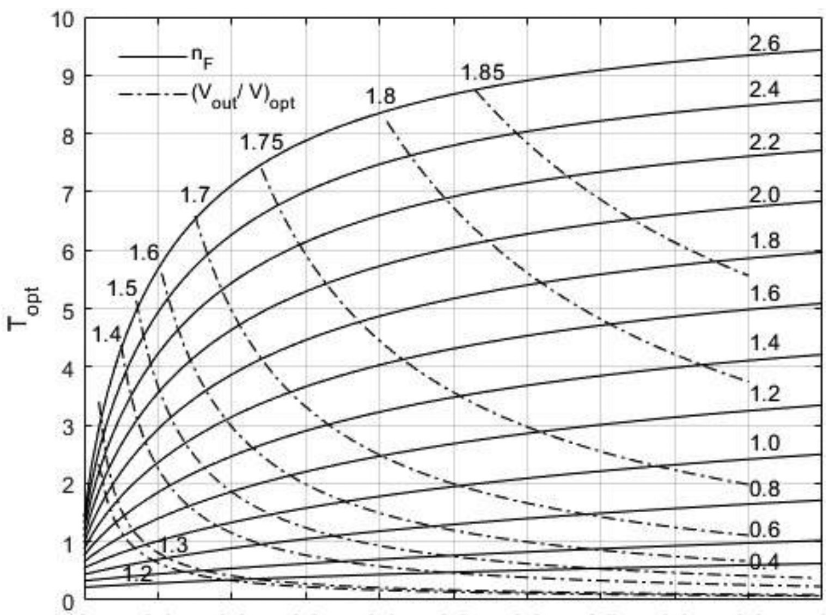

Fig. 2. Effect of losses and area ratio on the optimum diameter of the propulsor

The physical meaning of the optimal value of n_S and its variation is as follows. Given F and v, an increase in n_S results in the need to reduce the velocity v_{out}, but to a lesser extent. At the same time Q increases and the pressure p_v decreases due to the reduction of the kinetic energy of the water flow. When n_S decreases, the opposite is true. The minimum of the product p_vQ corresponds to the optimum n_S. However, the same change of the product p_vQ with the change of n_S at small values of the loss factor ζ occurs in the region of large values of n_S, and at large values of ζ – in the region of small values of n_S.

In the case when a ready-made axial fan is selected for a swimming vehicle, it is necessary to take into account also the following:

– when analyzing the hydrodynamic characteristics of axial ventilators it is necessary to take into account the case when the engine (gasoline or electric) is located in front of the impeller, and the wheel hub extends beyond the propulsion housing in the axial direction (Fig. 1), the dynamic pressure is calculated by the flow exit velocity, determined by the area of the blades (total area calculated by the wheel diameter, except for the area occupied by the wheel hub);

– in foreign catalogs, the dynamic pressure of axial fans is determined by the total area, i.e. the area covered by the wheel. The difference in static pressures, established by these methods, begins to noticeably affect the relative diameter of the hub $U > 0.4$ (the ratio of the diameter of the hub to the diameter of the fan). If this is not taken into account, the selected fan may not give the expected performance in a given design.

When selecting a fan from catalogs, attention should be paid to the following:

– whether the power specified in the characteristics is the fan power or the power consumed by the fan motor;

– whether the fan motor has a power reserve for starting loads, grille fouling and high-water temperatures.

The design of a fan propulsor is usually completed by a testing [11] and verification phase. Once the propulsor is designed, it is critical to test and verify the flow rate to ensure that it meets the intended specifications. Computational Fluid Dynamics (CFD) modeling [7, 12] is commonly used to predict water flow characteristics, but physical testing in controlled environments remains the gold standard for performance verification.

Conclusions:

1. Thus, the hydrodynamic performance of axial flow fans plays a key role in the propulsion of watercraft.
2. Calculation of the hydrodynamic characteristics of the axial fan it is advisable to carry out in the following order: selection of the structural scheme of the fan, flow kinematics, calculation of the aerodynamic characteristics of the fan in the operating range of variation of its performance, calculation of the aerodynamic characteristics of fans in their regulation.
3. For an axial fan should distinguish between static and dynamic pressure.
4. Consideration of the optimal relationship between pressure rise and flow velocity in the design of an axial fan is critical.
5. Power requirements for axial fan design are key in the design process.
6. Knowledge of water density is an important factor in axial fan design.
7. Efficiency is one of the most important parameters in designing an axial flow fan as it directly affects energy consumption and operating costs.
8. Propeller sizing method, when only the diameter of the fan is known, does not allow to judge the minimum possible power required to drive the fan.
9. Water flow testing and verification is critical to ensure that the axial flow fan meets the specified characteristics.

10. The optimum combination of all these components will ensure high performance and meet the designers' expectations for water jet propulsion quality.

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