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EFFECT OF 3D SHAPE OF PUMP-TURBINE RUNNER BLADE ON FLOW CHARACTERISTICS IN TURBINE MODE

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The effect of blade spatial profiling with the help of tangential blade lean of Francis pump-turbine runner with heads up to 200 m on the flow structure and energy characteristics was numerically investigated. A flow part model of Francis pump-turbine of the Dniester pumped storage plant was adopted as original version. Two new blade systems were designed, which differed from the original version by mutual position of cross-sections in tangential direction: with positive and negative lean, while the shape of the cross-sections themselves remained unchanged. Modeling of the viscous incompressible flow in calculation domain, which contains one channel of the guide vane and the runner, for three variants of flow parts, was performed using the IPMFlow software based on numerical integration of the Reynolds equations with an additional term containing artificial compressibility. To take into account the turbulent effects, the SST differential two-parameter turbulence model of Menter is applied. Numerical integration of the equations is carried out using an implicit quasi-monotonic Godunov scheme of second order accuracy in space and time. The study was carried out for models with runner diameter of 350 mm in a wide range of guide vane openings at reduced rotation frequencies corresponding to the minimal, design and maximal heads of the station. A comparison of pressure fields and velocity vectors in the runners, pressure graphs on runner blades, distribution of velocity components at inlet to a draft tube, and efficiency of three variants of flow parts are presented. It was concluded that calculation domain with the new RK5217M2 runner with negative tangential lean has the best characteristics. An experimental study of three runners on a hydrodynamic stand are planned.

Keywords: *runner blade, Francis pump-turbine, flow part, tangential lean, numerical study, spatial flow, flow structure.*

In the European Union, in 2021, the share of green electricity produced by renewable energy sources (RES), in particular hydroelectric power plants, wind turbines and solar power plants, was 38%, nuclear power plants – 25%, and fossil fuels – 37%. In general, the share of "clean" energy in the world in 2021 reached 37.88% [1]. In the same year in Ukraine, renewable energy produced 14.7%, in particular 6.7% from hydroelectric power plants and pumped storage plants, 8% from solar power plants and wind turbines, 55% from nuclear power plants, and 30.3% from thermal power plants [2].

With the post-war reconstruction and the development of Ukraine's energy system, the determining importance for meeting the needs for highly maneuverable balancing and accumulative capacities should be given to the pumped storage plants. Among modern electricity balancing technologies, pumped storage plants are the most efficient and the most widespread in the world. They make up almost 94% of all balancing capacities [2]. Hydroelectric power plants can not only produce electricity as hydropower plants, but also consume excess electricity (for example, produced by nuclear power plants at night or wind turbines and hydroelectric power plants during the day), pumping water from lower to upper reservoir, due to which they balance the load schedule [3].

The total capacity of hydropower pumped storage plants in the world is approximately 170 GW, in particular, more than 53 GW in Europe (Germany – 5.7 GW, France – 5.7 GW, Italy – 4 GW). The installed capacity of hydropower plants in the world increased by 25 GW in 2021, including 6.3 GW of new pumped stor-

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age capacities [4]. According to the International Hydropower Association, their capacity is expected to double by 2030 [2].

In Ukraine today, the capacity of the pumped storage plants is only 2.0 GW. In Ukraine, there are sites for new promising medium (capacity up to 300 MW) and large (capacity 500–1300 MW) pumped storage plants with a preliminary total capacity of 10 GW. These are what can ensure the balancing of the Unified Energy System of Ukraine with its further development [2].

Hydraulic units with Francis turbine are installed at almost all pumped storage plants. The runner blades of such turbines have a significantly spatial shape, which greatly complicates the design and further optimization. A significant number of design and optimization methods have been collected, suitable both for large Francis turbines [5], Kaplan turbines [6], and for microturbines [7]. Optimizing energy indicators requires the presence of limited number of variable parameters describing the geometric and mode characteristics of a hydraulic machine. To describe the shape of blades, both simplified methods using high-order polynomials for individual sections [8] and complex, fully three-dimensional methods are used, for which the geometric description does not depend on the mode parameters [9]. The initial approximation of geometry for optimization according to the given initial parameters of turbine can be obtained both with the help of analytical methods and with the use of commercial or open-source software [10].

Studies of recent years show that one of the promising directions for increasing the efficiency of power machines is the use of tangential and axial lean of runner blades [11, 12]. This approach affects structure of the flow [13], pulsating characteristics of the flow [14], as well as energy indicators of the flow part.

In recent years, the frequency of runner rotation has been increasingly adopted as optimization parameter [15], including as part of multiparameter optimization with a change in geometric characteristics [16, 17].

Optimization of the energy characteristics of turbine can be carried out both for one [17] and for two or more mode points [18], including for both turbine and pump modes [19]. The goal of optimization can also be cavitation coefficient [20] and erosion wear parameters [21].

There are different approaches to determine the target optimization parameters: more common is numerical simulation of flow in calculation domain of turbine, which includes only the runner and the guide vane [21], but some researchers insist on the mandatory inclusion of the stator columns and the draft tube in the calculation domain [20].

The study of fluid flow in pump-turbines can be carried out both numerically and experimentally [22, 23]. The numerical modeling of the flow allows researcher to significantly save the time and cost of designing pump-turbine elements, obtain and analyze the flow structure in any cross-section of elements of flow part. The experimental studies are a more reliable way to verify the obtained energy, cavitation and pulsation characteristics. In addition, they may reveal phenomena that, for various reasons, were not detected during numerical experiments. This especially applies to non-stationary phenomena, which sometimes require careful selection of numerical modeling modes [24]. In most cases, experimental studies are conducted to confirm calculated characteristics of developed or optimized turbine [22, 25].

The paper presents results of design and numerical study of the effect of tangential lean of runner blades of Francis type pump-turbine at heads of up to 200 m.

As shown above, one of the promising directions for increasing the efficiency of Francis pump-turbines is the spatial profiling of the runner blades with the help of the tangential lean. With this approach, shape of edges and mutual position of cross-sections in the tangential direction changes due to the shift of the hub cross-section in direction of rotor rotation (positive lean) or in opposite direction (negative lean). The shape of sections itself remains unchanged. The shift can be carried out linearly relative to the height of runner at the pressure edge or according to a polynomial law. This approach ensures obtaining the optimal shape of the blade, which best interacts with the flow both in area of the leading edge and in channels of the runner and when forming the flow at inlet to the draft tube.

As an object of the research, a model of the highly efficient Francis pump-turbine ORO170/5217, which was developed at the IMEP and implemented on the first 4 hydrounits of the Dniester PSP, was adopted. In the RK5217 runner, the suction edge is located in radial plane, the angle of inclination of pressure edge in tangential direction is 10º in the direction of runner rotation. In order to study the effect of spatial profiling of blades on the energy characteristics of the flow part in turbine mode, two modifications of the runner, named RK5217M and RK5217M2, were designed using the tangential lean. In the RK5217M, the angle of inclination of the pressure edge is 45°, the hub section is shifted in the direction of rotor rotation by 12.8° relative to the original version, all others are linearly dependent on the height of runner. In the RK5217M2, the angle of inclination of the pressure edge is -45°, accordingly, the hub section is shifted to the side opposite to rotation by 18.2° relative to the original RK5217. Computer models of the original and modified blades are shown in Fig. 1.

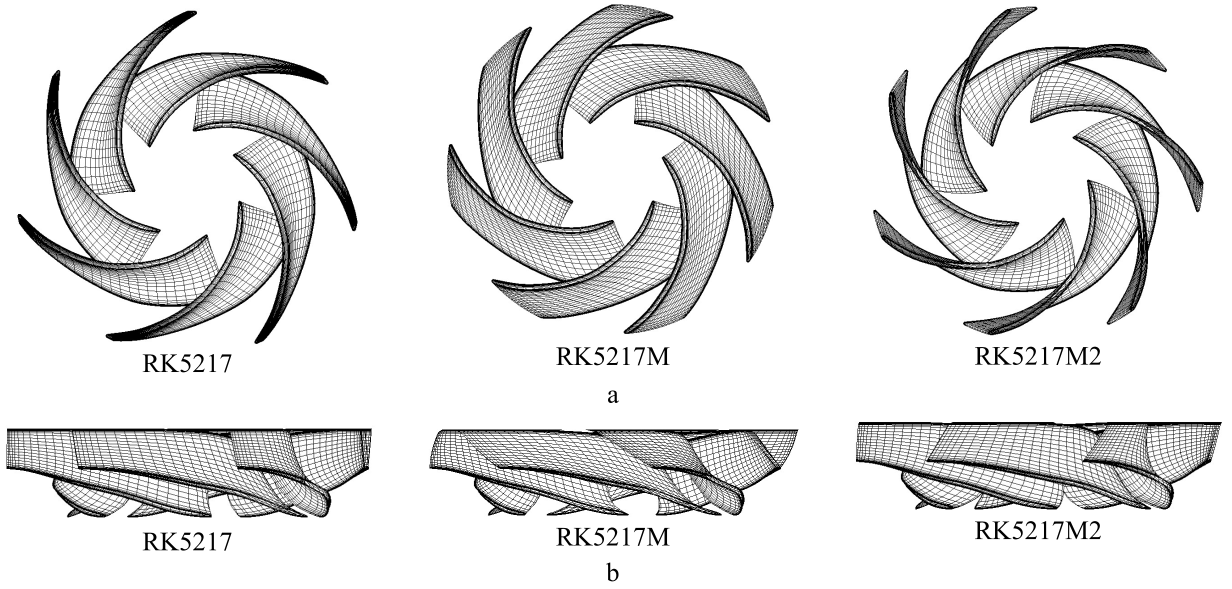


Fig. 1. Computer model of three variants of runners:
a – top view; b – side view

The numerical studies of the effect of spatial profiling of runner blades on the flow structure and energy characteristics of flow parts in turbine mode were out using the *IPMFlow* software, developed by IMEP.

Modeling of a viscous incompressible fluid flow in the pump-turbine models is performed on the basis of numerical integration of the Reynolds equations with an additional term containing artificial compressibility.

Turbulent effects were modeled using Menter's differential two-parameter model (SST). Numerical integration of the equations was carried out using an implicit quasimonotone Godunov scheme of the second order approximation in space and time.

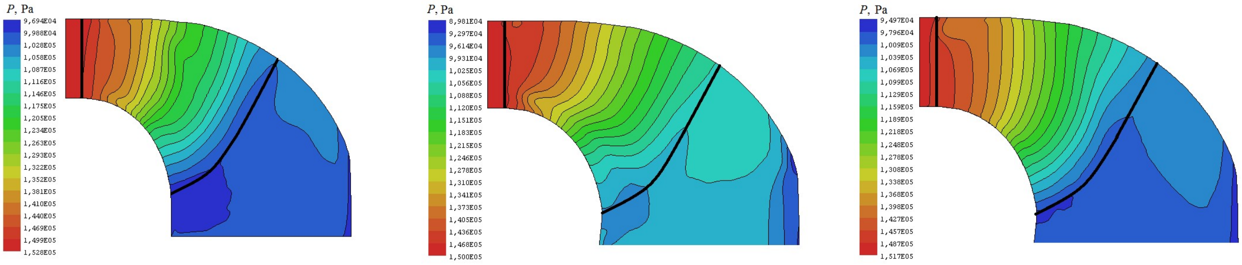
The calculations were made for models with a diameter of 350 mm, which corresponds to the dimensions of the studied models on the IMEP ECS-30 hydrodynamic stand. The calculation domain contains one channel of the guide vane and one channel of the runner. The normalized hexagonal calculation mesh had $72\times72\times80=414{,}720$ and $80\times80\times100=640{,}000$ elements in the channel of the guide vane and the runner, respectively, with thickening near the walls. The value of the indicator y^+ did not exceed 10.

The study was conducted at head of 6 m (which was equal to the head of the planned experimental tests) at three values of the reduced rotation frequency: 95; 91; 85 min^{-1}, which correspond to the minimum, design and maximum heads at the Dniester PSP; in wide range of openings of guide vane: 12; 16; 20; 24; 28 and 34 mm.

In Fig. 2 is shown the pressure distribution in the middle meridional cross-sections of the runner channels for three variants of the 5217 series runner at rotation frequency of 91 min^{-1} and opening of guide vane a_0=24 mm.

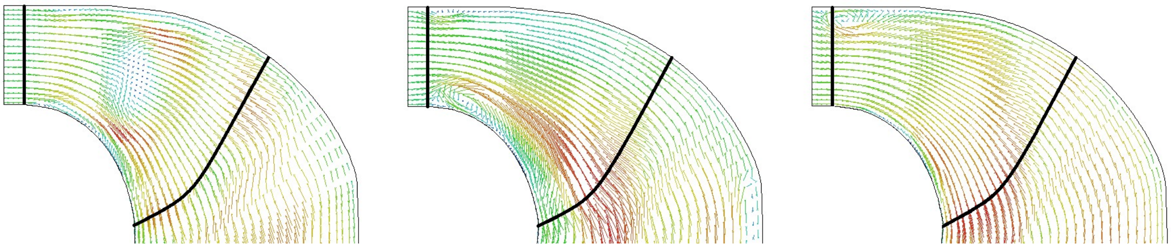
At almost all values of head and guide vane openings, the most uniform distribution of pressure isolines is observed in the RK5217M2 channel with negative tangential lean of blades. The lean in this runner is expected to have some effect on the pressure fields in area of pressure edge near the hub, but leads to the most favorable distribution in peripheral zone and along entire height of suction edge. The least uniform pressure distribution is observed in the RK5217M channel with positive circular lean of blades.

Fig. 3 shows the distribution of velocity vectors in the middle meridional sections of the runner channels for three variants of the 5217 series runner at rotation frequency of 91 min^{-1} and opening of guide vane a_0=24 mm.



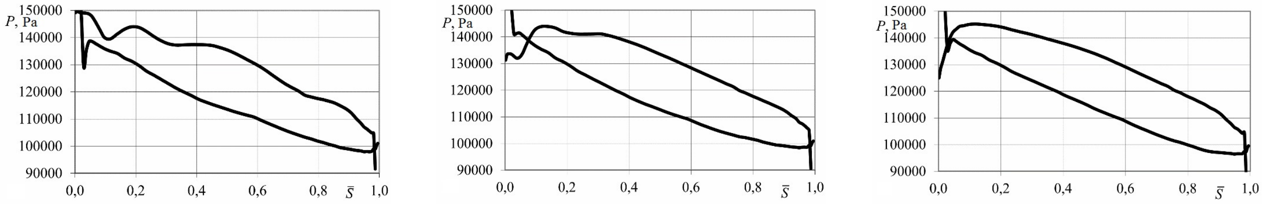
RK5217 RK5217M RK5217M2

Fig. 2. Pressure distribution in the middle meridional section at design head (n_I'=91 min^{-1}) and opening of the guide vane a_0=24 mm in turbine mode



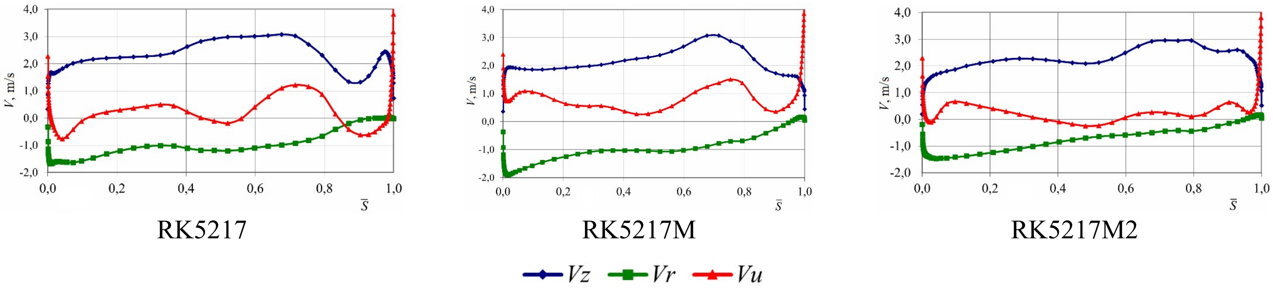
RK5217 RK5217M RK5217M2

Fig. 3. Distribution of velocity vectors in the middle meridional section at nominal head (n_I'=91 min^{-1}) and opening of the guide vane a_0=24 mm in turbine mode



RK5217 RK5217M RK5217M2

Fig. 4. Distribution of pressure in the middle cross-sections of blades of three variants of the 5217 series runner at design head (n_I'=91 min^{-1}) and opening of guide vane a_0=24 mm in turbine mode



RK5217 RK5217M RK5217M2

Fig. 5. Distribution of the axial V_z, radial V_r and circular V_u components of velocity at the exit from runner at nominal pressure (n_I'=91 min^{-1}) and guide vane opening a_0=24 mm

The most uniform distribution of vectors along the runner channel is observed in flow part with the RK5217M2. Unlike the other two runners, there are practically no secondary flows in it, except for zone of pressure edge near hub, and the most uniform flow is formed at the exit from runner in the front of draft tube.

Fig. 4 shows the distribution of pressure graphs on the middle sections of blades of three variants of the 5217 series runner with a rotation frequency of 91 min^{-1} and an opening of guide vane a_0=24 mm.

Analysis of the graphs leads to conclusions that the most uniform graphs at all values of pressures and guide vane openings are observed for the RK5217M2. First of all, the pressure distribution in the area of leading edge on pressure side is improved and equalized relative to the original version of RK5217. Thus, the use of negative lean of blades leads to a better work of pressure and, as a result, should increase the efficiency of the flow part. The pressure graphs on blades of the RK5217M with positive lean are the most uneven among other runners of the 5217 series, especially at smaller values of guide vane openings. A characteristic feature of all runners of the series is the small work of pressure in the inlet region of cross-sections at small values of guide vane opening. With an increase in flow rate, the appearance of the graphs of all runners improves.

In Fig. 5 the distribution of the axial Vz, radial Vr and circular Vu components of speed at the exit from runner at rotation frequency $n_I'=91$ min^{-1} and guide vane opening $a_0=24$ mm is shown.

In almost all modes studied, the use of negative tangential lean of the RK5217M2 blades led to a noticeable improvement in the distribution of speed components: the axial and circular components at the exit from runner change by width more smoothly, which creates better conditions at the inlet to draft tube. Such changes in the structure of flow should lead to an increase in overall efficiency in wide range of operation of the hydraulic unit in generator mode. At the exit from the RK5217M with a positive lean of blades, the most uneven distribution of both axial and circular speeds among the 5217 series runners is observed.

In Fig. 6 is shown the comparison of dependences of the efficiency* of the calculation domain on values of the guide vane openings for three variants of the 5217 series runners at the minimum, design and maximum heads in turbine mode.

As can be seen, at all values of heads and guide vane openings, the best energy indicators were obtained in flow part with the RK5217M2, which uses blades with negative tangential lean and which has the best characteristics of flow structure in elements of the flow part (Fig. 2–5). The level of its calculated efficiency significantly exceeds the indicators of the flow part with the original version of highly efficient RK5217, which has been installed and successfully operated for many years at hydrounits no. 1–4 of the Dniester PSP. In the calculation domain, the lowest level of efficiency was obtained with the RK5217M with positive tangential lean. Thus, by the method of numerical experiment, it was established that the use of tangential blade lean of pump-turbine runner at heads of up to 200 m allows to significantly influence the flow structure and the energy characteristics of flow part, which in turn made it possible to design a runner with improved characteristics. In order to confirm the numerically obtained results of the study of the effect of leans on the energy characteristics of pump-turbines in generator mode, experiments on a hydrodynamic stand are planned.

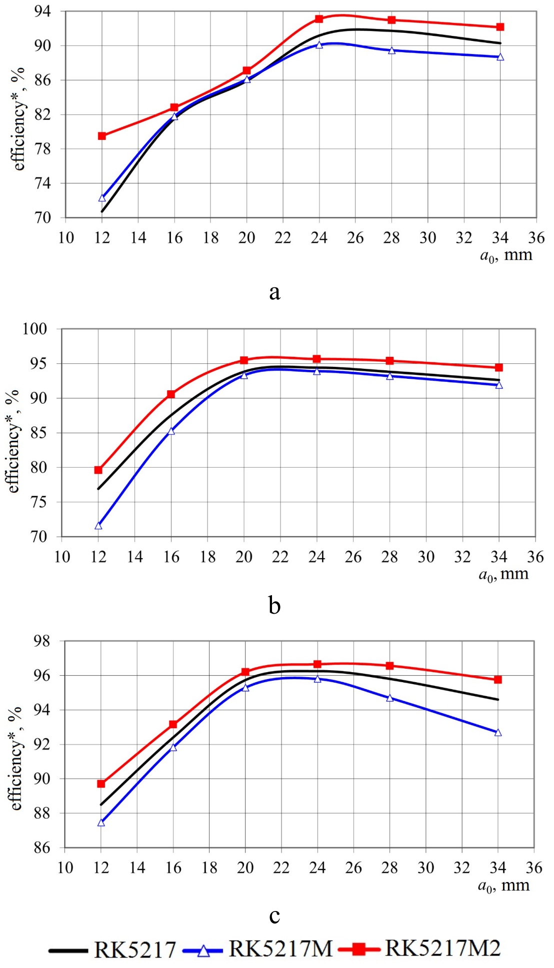


Fig. 6. Dependencies of the estimated efficiency* on the guide vane opening in the pump-turbine model with three variants of runners in turbine mode:
a – minimum head ($n_I'=95$ min^{-1}); b – design head ($n_I'=91$ min^{-1}); c – maximum head ($n_I'=85$ min^{-1})

Conclusions

1. A method of spatial profiling of Francis pump-turbine runner blades is proposed, which is based on the application of blade lean in tangential direction and allows changing flow structure in flow part of the Francis pump-turbine and its energy characteristics.

2. The dependence of the flow structure and the efficiency values in flow parts on tangential lean of pump-turbine runner blades in turbine mode was numerically determined.

3. The negative tangential lean of the RK5217M2 blades led to an improvement in the flow structure and the efficiency level, while the positive lean in the RK5217M led to their deterioration.

4. According to the results of numerical studies, it was established that the flow part model with the RK5217M2 in terms of energy performance exceeds the original version with the RK5217, therefore, after experimental verification on a hydrodynamic stand, it can be recommended for implementation in the project of hydrounits no. 5-7 of the Dniester PSP.

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Вплив просторової форми лопатей робочих коліс насос-турбіни на характеристики потоку в турбінному режимі

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У чисельний спосіб досліджено вплив просторового профілювання за допомогою колових навалів лопатей радіально-осьових робочих коліс насос-турбін з напорами до 200 м на структуру потоку і енергетичні характеристики. Як вихідний варіант прийнято модель проточної частини радіально-осьової насос-турбіни Дністровської ГАЕС. Спроєктовано дві нові лопатеві системи, що відрізнялися від вихідного варіанта взаємним розташуванням розрахункових перерізів у коловому напрямі: із позитивним і негативним навалом, при цьому форма самих перерізів залишалася незмінною. Моделювання нестисливої течії в'язкої рідини в розрахункових областях, що містили по одному каналу напрямного апарата і робочого колеса, трьох варіантів проточних частин виконано за допомогою програмного комплексу IPMFlow на основі чисельного інтегрування рівнянь Рейнольдса з додатковим членом, що містить штучну стисливість. Для врахування турбулентних ефектів застосована диференціальна двопараметрична модель турбулентності SST Ментера. Чисельне інтегрування рівнянь проводиться з використанням неявної квазімонотонної схеми Годунова другого порядку точності за простором і часом. Дослідження проведено для моделей з діаметром робочого колеса 350 мм в широкому діапазоні відкриттів напрямного апарату при приведених частотах обертання, що відповідають мінімальному, номінальному і максимальному напорам на станції. Наведено порівняння полів тиску і векторів швидкості в каналах робочих коліс, епюр тиску на лопатях коліс, розподіл компонент швидкості на вході у відсмоктувальну трубу, а також ККД трьох варіантів проточних частин. Зроблено висновок, що найкращі характеристики має розрахункова область з новим робочим колесом РК5217М2 з негативним втулковим коловим навалом. Заплановано експериментальні дослідження трьох коліс на гідродинамічному стенді.

Ключові слова: *лопать робочого колеса, радіально-осьова насос-турбіна, проточна частина, коловий навал, чисельне дослідження, просторова течія, структура потоку.*

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