

Investigation into Pump Mode Flow Dynamics for a Mixed Flow PAT with Adjustable Runner Blades

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Abstract: The adoption of Pump as Turbines (PATs) both in small scale hydroelectric plants and water supply systems has brought different advantages, the most recognized being cost-effectiveness as compared to other hydroturbines. However, due to their lack of flow control ability, their intolerance to off-design operations constitutes one tough shortfall. Moreover, this technology's newness leads to PAT flow dynamics still being ununderstood even to date. Therefore, this study intends to numerically investigate the mixed flow PAT's pump mode flow dynamics for five operating conditions expanding from optimum ($1Q_{BEP}$) to deep part-load ($0.41 Q_{BEP}$) conditions. Moreover the effect of runner blade angle on the same has been investigated where three angles namely -2° , 0° , and 2° have been considered. PAT flow stability was found to deteriorate as the flow decreased, where associated pressure pulsation level worsened at different flow zones. In addition, the blade angle increase led correspondingly increasing flow unsteadiness and pressure pulsation levels, where the pulsation frequencies from rotor-stator interactions were dominant for most of flow zones. This study's findings are of a crucial importance to both scientific and engineering communities as they contribute to thorough understanding of PAT flow dynamics.

Keywords: Pump as turbine, Flow unsteadiness, Pressure pulsation, Blade angle, Numerical simulation

1. Introduction.

The usage of hydraulic pumps in energy generation systems, which is generally known as the pump as turbine (PAT) technology, has brought different advantages that mainly improve the feasibility of energy supply projects at remote localities far from the electric grid reach [1-3]. In addition, this technology can be used to alleviate pressure levels within water distribution networks, at the same time producing usable electric power, where resulting energy savings could be in the considerable amounts [4-7]. As mentioned through different published works, PATs outperform similar microscale hydroturbines by one main feature: Cost-effectiveness [8-12]. Otherwise, PAT-associated technical shortfalls such as the intolerance of off-design operating conditions or the widely known lack of accurate conversion means between pumping and turbinning performance characteristics would hinder its adoption [13-15]. In line with this, different studies have been continuously carried out; addressing the already available technological issues within PAT's both operating modes [15-17]. In this respect, in addition to PAT performance prediction studies [18-22], investigations into PAT flow unsteadiness under off-design operating conditions and respective flow structure formation mechanism have been carried out, which in a long run should lead to the improvement in terms of PAT's well-balanced performance in both its operating modes. Among the recently published details on the same theme, a drop in PAT's mechanical efficiency has been recorded under part-load conditions, as a result of the occurred flow detachment and swirling flows within the machine flow channels [23]. In their study, PAT's BEP Flow rate (Q_{BEP}) and Head (H_{BEP}) under turbine mode have been found to overtake the pump mode by 27% and 41% respectively. In the same respect, Shen et al. [24] using the Entropy generation theory for PAT's pump mode, pointed out the impeller and diffuser to be the main flow domains with higher mechanical energy losses, accounting for 35.32% to 55.51% and 32.61% to 20.42 respectively. These ones were found to be closely associate with the machine hump characteristics and became more and more pronounced as the machine flow rate decreased. It's also worth noting that the increase in tip clearance only worsened the turbulent dissipation at the impeller blade's tip area, leading to poorer machine performance. Both Ghorani et al. [25] and Lin et al. [26] used the above entropy theory to analyze PAT energy losses under turbine operating mode and dissect the most influential flow zones. Both studies have again converged to a realization that the runner and the volute casing flow zones constitute the flow zones of higher hydraulic losses, where for instance, Ghorani et al. [25] highlighted that 50% of energy dissipation could take place within the runner itself, owing to the occurred complex flow phenomena such as inter-blade back flow from the emerged inter-blade vortices and flow separations at blade leading and trailing edges, among others. Trying to improve the PAT performance under part, full, and over-load conditions, Ntiri et al. [27], utilizing an optimization scheme, managed to improve the machine's efficiency by 23.7%, 11.5%, and 10.4% for the above mentioned operating conditions respectively. Subsequently, the inter-blade flow field for the optimized model attained a far better state, where under part-load for instance, runner inter-blade pressure distribution was uniform and vortices were considerably weakened. In another study [28], the attempt was done to optimize both the turbine and pump mode under design flow conditions for one rotational speed and the obtained results were quite satisfying. After a realization that PAT back-curved blade design does not match the turbine mode where flow separations at the blade leading edge (BLE) is almost unavoidable, Wang et al. [29] designed a special runner with forward-curved blades for use under PAT's turbine mode, where the increase in blade inlet angle led to higher BEP flow conditions. Compared to the normal design, the new design exhibited a 13.2% improvement in terms of the machine best efficiency. The clocking effect on PAT performance considering a diffuser kind of pump, has been investigated by Hongyu et al. [30], while the effect of rotating speed was studied by Tahani et al. [31]. The same parameter was analyzed by Han et al. [32], but now studying the effect of the same on tip leakage vortex, considering a mixed flow PAT in pump mode, where it was generally found that the increase in rotating speed worsens the tip leakage vortex. Speaking of both the axial and mixed flow PATs, which constitute the less discussed types in this field, a number of results have been published where the focus is mostly put on tip leakage vortex dynamics [33, 34].

In order to get a thorough understanding of PAT's turbine mode operations and associated flow dynamics, especially considering a mixed flow PAT, one would have to investigate its pump mode characteristics and the effect of different parameters on the same. In this respect, the present article presents a numerical study that investigates the local flow structures formation mechanism and their eventual changes as the machine flow conditions change from part-load to full-load, primarily keeping the focus on PAT's pumping mode. In addition, an attempt has been done to also investigate the effect mechanism of runner blade angle on both local

flow dynamics and associated pressure pulsation characteristics, where three blade angles, namely -2° , 0° , and 2° have been considered.

2. PAT physical model and numerical method

2.1. PAT geometric model

In this study, the investigated PAT model is of mixed flow type with a high specific speed ($N_q=163.9$), composed of four main components, namely the inlet pipe, the runner, the diffuser, and outlet pipe. The model's geometrical parameters have been shown in Table 1. The number of runner blades and guide vanes are 3 and 8 respectively. For numerical simulations, the investigated computational domain, englobing the above mentioned four components, has been built using a geometrical modeling software, Unigraphix NX8.0. Fig.1 shows the utilized computational domain and components.

Table1. PAT model geometric parameters

Parameter	Symbol	Value
Runner Specific Speed (min^{-1})	N_q	163.9
Runner Inlet Dia. (mm)	D_{in}	300
Runner Dia. (mm)	D_R	320
Runner Blades Number (-)	Z_R	3
Guide Vane outlet Dia. (mm)	D_O	350
Guide Vanes number (-)	Z_G	7
Blade tip clearance (mm)	δ	0

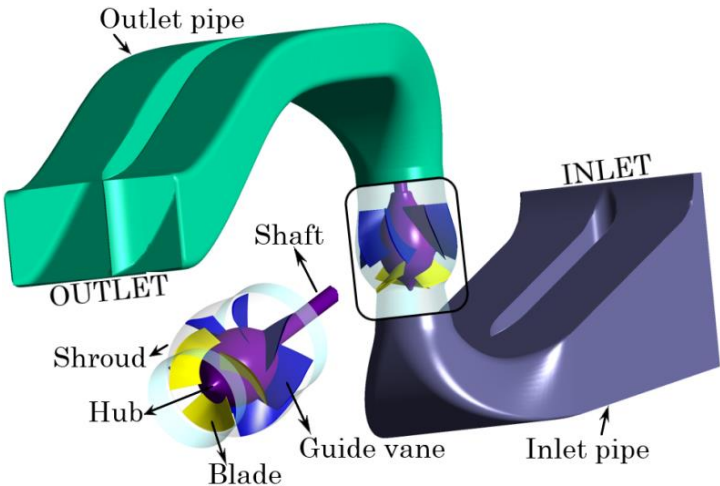


Fig.1 PAT computational domain and components

In this respect, performed numerical simulations consider the water flow entering the computational domain from the inlet zone of the inlet pipe, then gets whirled through the runner's inter-blade and inter-guide vane flow channels before passing through the outlet pipe, heading towards the domain's exit zone. This trajectory is imitated from the experimental process where the flow followed the same direction. The investigated model, unlike many recently published works, presents no runner blade tip clearance. All the experiments have been conducted, respecting the international standards for hydraulic machinery testing, i.e. the International Electrotechnical Commission standards or IEC[35]. Experimental tests on the machine's external performance characteristics have been carried out for a wide range of flow conditions ranging from deep part-load flow conditions, through part-load, upper part-load, full load, all the way up to over-flow flow conditions.

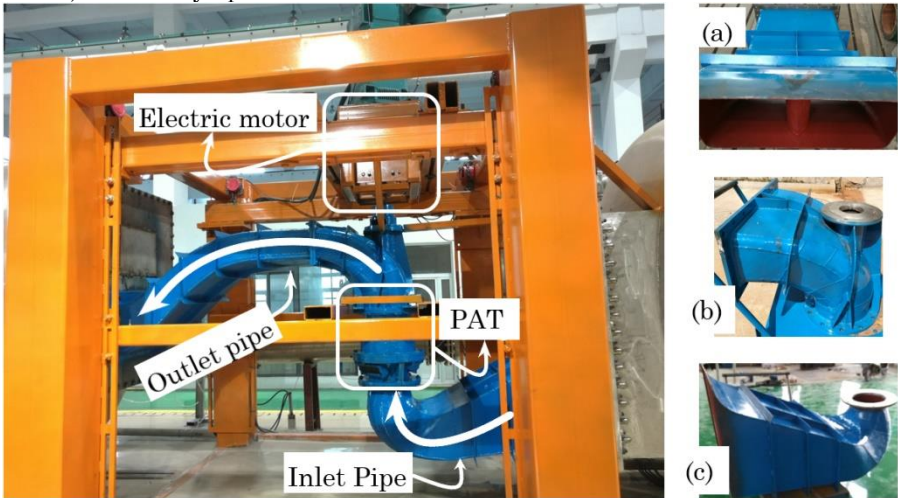


Fig.2 Experimental test rig and components (a) outlet pipe, (b) shaft zone, and (c) inlet pipe

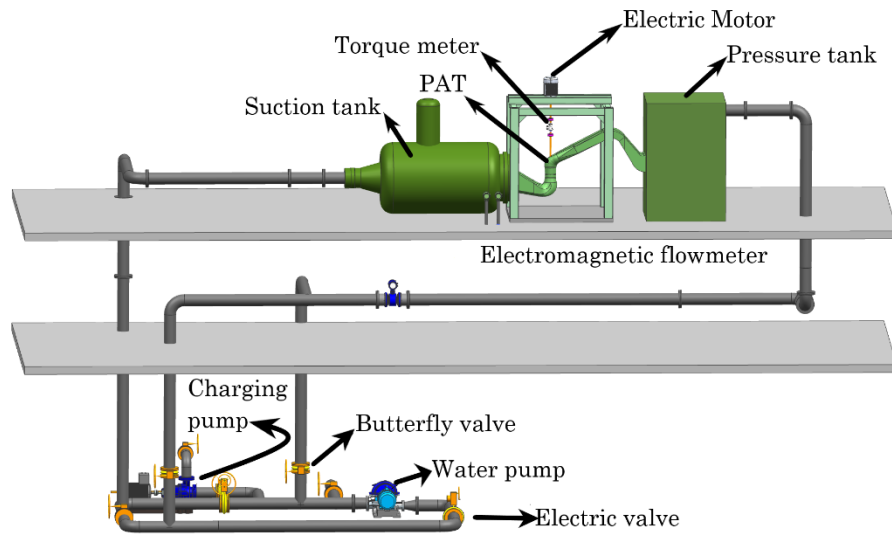


Fig.3 Schematic diagram of the PAT testing system

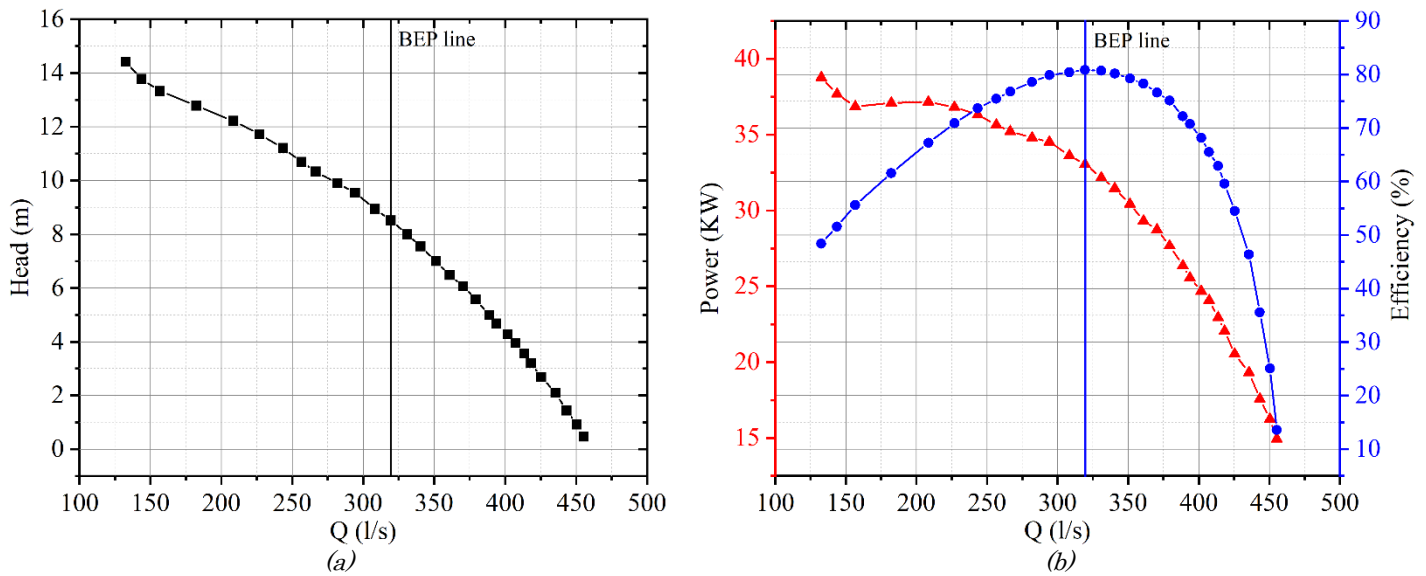


Fig.4 Machine external performance characteristics (a) H - Q curves (b) η - Q curves

Fig.2 shows the utilized experimental testing rig and some components of the tested machine components, while the schematic view of the testing system is given in Fig.3. In Fig.4, considering the runner blade angle of -2° , experimentally found machine external performance characteristics in terms of variations of developed head (H), Efficiency (η), and shaft power (P) are presented, where the vertical line around the mid-zone of each graph is the BEP line. It can be seen that with a continuous flow discharge increase, both the developed Head and shaft Power decreased, whereas the Efficiency first increased to BEP point before dropping for farther flow conditions. Having conducted experimental tests on a wide range of flows, the best efficiency operating point was reached under 319.49 l/s flow rate conditions, and is characterized by the developed head of 8.52m, the shaft power of 33.04kW, and machine overall efficiency of 80.84%.

2.2. Numerical method and validation

2.2.1. Computational grid generation

In line with this study's main objectives, numerical simulations are conducted to investigate the machine flow dynamics and their evolution mechanism responsible for different performance characteristics as shown in the above sections. As a preparatory phase, a computational grid has to be generated out of the whole computational domain of concern. In this study, the Ansys ICEM software has been used to generate computational grids of the PAT model's four components, where the hexahedral scheme has been utilized for most of flow domains. A comparatively fine mesh has been generated at flow zones of high importance such as the runner blades and guide vanes or any other flow deflection zones (turns or corners), where the global maximal y^+ value is less than 50.

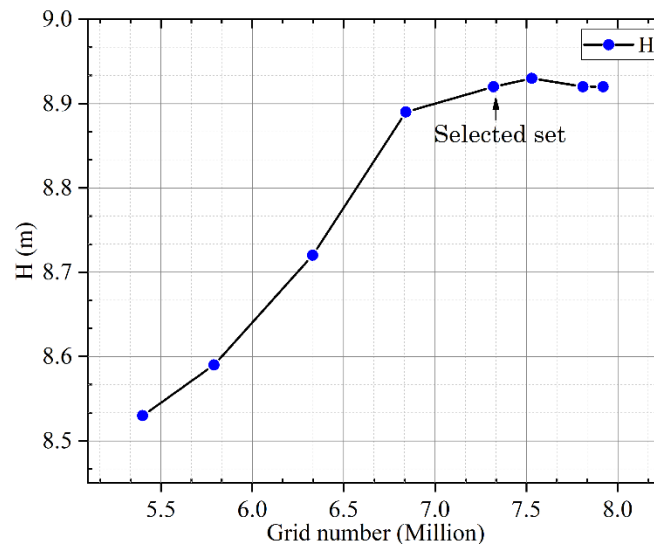


Fig.5 Grid independence test.

Table 3 Selected grid details for each PAT components

Components	Inlet pipe	Runner	Guide vanes	Outlet pipe
Grid type	Hexahedral	Tetrahedral	Tetrahedral	Hexahedral
Grid number (Million)	1.2	3.41	3	1.31
Grid Quality	4.3	2.8	3.2	4.1

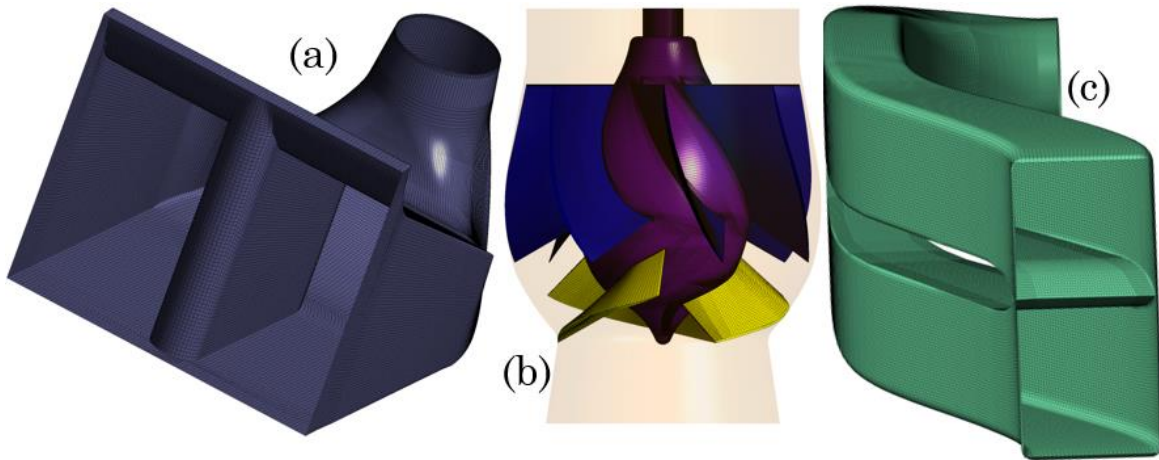


Fig.6 Selected grid for different model components (a) inlet pipe, (b) guide vane and runner, and (c) Outlet pipe

For the sake of eliminating the possibility of the utilized grid number influencing numerical simulation results, a grid independence test has been conducted where eight different grid numbers ranging from 5.4 to 9.7 million have been tested. Steady state numerical simulations have been conducted on each grid set under similar boundary conditions, after which, resultant performance characteristics in terms of Head (H) have been compared. As shown in Fig. 5, the developed head has continuously increased with the grid number for all grid numbers below 8.92 million while it almost stabilized for grid numbers above. Therefore, in line with the available computational resources, the grid number of 8.92 million has been chosen for further simulations. Table 3 shows the details of the selected grid at different components of the investigated PAT model, while Fig.6 gives a quick glance on the generated mesh at all flow zones.

2.2.2. Numerical simulation scheme and validation

The 3D numerical simulations on PAT flow under different operating conditions have been carried out using the CFD's commercial code Ansys CFX. This code, using the Shear Stress Transport turbulence model, solves the Reynolds Averaged Navier-Stokes (RANS) equations, that themselves guide the motion of incompressible as well as compressible fluids.

$$\frac{\partial u_i}{\partial x_i} = 0 \quad (1)$$

$$\rho u_j \frac{\partial}{\partial x_j} (u_i) = \rho F_i - \frac{\partial p}{\partial x_i} + \mu \frac{\partial^2 u_i}{\partial x_j \partial x_j} \quad (2)$$

The investigated system, being the case of an application of hydraulic pumps for both the pumping and energy generation purposes, uses natural water at ambient temperature (25°C), thus classifying as an incompressible fluid system. The continuity and RANS equations for incompressible flows are written in their Cartesian form in Equation 1 and 2. In these expressions, ρ , μ , p ,

and F_i are the fluid density, viscosity, pressure, and body forces. The SST $k-\omega$ turbulence model, being developed by Menter in 1993 [36, 37], combines the advantages of two former and commonly used computational models, namely the $k-\epsilon$ and $k-\omega$ models [38, 39], leading to an improved ability to quite reasonably predict flow dynamics within the wall vicinities as well as the fluid bulk far from the wall. This model is also known for its good performance especially when dealing with simulation cases with adverse pressure gradients and flow separations [40]. The associated expressions of turbulent kinetic energy, specific dissipation rate, and kinematic eddy viscosity are given in Equation 3, 4 and 5.

$$\frac{\partial k}{\partial t} + u_j \frac{\partial k}{\partial x_j} = P_k - \beta^* k \omega + \frac{\partial}{\partial x_j} \left[(\nu + \sigma_k \nu_T) \frac{\partial k}{\partial x_j} \right] \quad (3)$$

$$\frac{\partial \omega}{\partial t} + u_j \frac{\partial \omega}{\partial x_j} = \alpha S^2 - \beta \omega^2 + \frac{\partial}{\partial x_j} \left[(\nu + \sigma_\omega \nu_T) \frac{\partial \omega}{\partial x_j} \right] + 2(1 - F_1) \sigma_{\omega,2} \frac{1}{\omega} \frac{\partial k}{\partial x_i} \frac{\partial \omega}{\partial x_i} \quad (4)$$

$$\nu_T = \frac{a_1 k}{\max(a_1 \omega, S F_2)} \quad (5)$$

During the 3D numerical simulation of PAT's flow under different conditions, targeting a quick and smooth solution convergence, steady state numerical simulations were first conducted, the results of which served as the initial conditions for transient numerical simulations. For each simulation session, the values of utilized boundary conditions were picked from the above presented experimental data. Though experimental tests have been conducted on a wide range of flows (Q) and blade angles (φ), the present study only focuses on a small number of flow conditions while considering a single blade angle ($\varphi = -2^\circ$). The investigated operating conditions range from part-load to full load ($1Q_{BEP}$), where four cases have been considered for part-load operating zone, namely the deep part-load ($0.41 Q_{BEP}$), part-load ($0.65 Q_{BEP}$), and upper part-load ($0.83 Q_{BEP}$ and $0.92 Q_{BEP}$) conditions. However, in order to investigate the effect of blade design on PAT flow instability, the runner blade angle was increased for two successive values, namely $\varphi = 0^\circ$ and $\varphi = +2^\circ$, considering the most unstable flow conditions. Throughout the simulation process, the whole PAT computational domain was divided into stationary and rotating frames, between which different interfaces have been used. For instance, the frozen rotor-stator interface type has been utilized at both the runner inlet and outlet flow zones for steady state numerical simulations, while the transient rotor-stator type was adopted at same flow zones under transient numerical simulations.

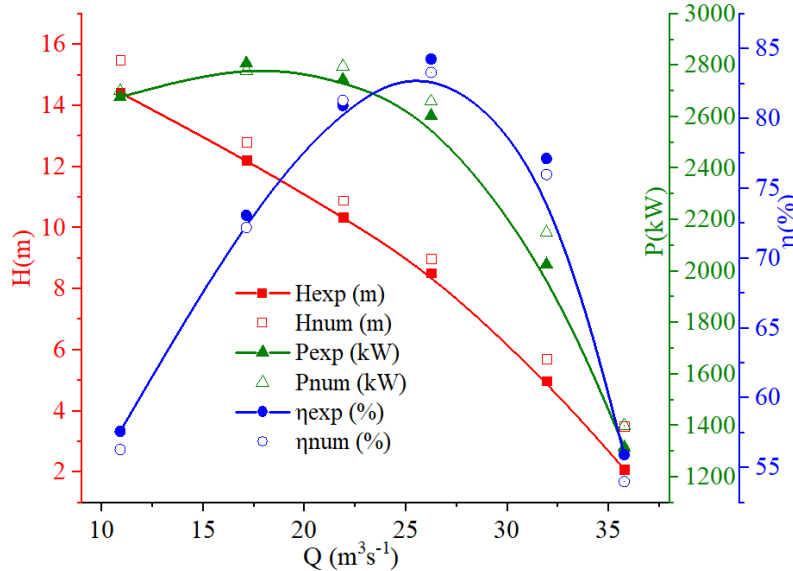


Fig.7 Comparison between numerical and experimental results

$$H = \frac{P_2 - P_1}{\rho g} \quad (6)$$

$$P = \frac{2\pi n T}{60} \quad (7)$$

$$\eta = \frac{\rho g Q H}{P} \times 100 \quad (8)$$

For other interfaces linking stationary components, the general grid interface (GGI) has been used. In addition to P_{in} and Q_{out} as inlet and outlet boundary conditions, the non-slip wall condition has been imposed at all other walls of the concerned computational domain. High resolution scheme has been chosen for both the advection scheme and turbulence numerics. For each transient solution, the timestep Δt has been set as the time required for the runner to rotate for 1 degree, which means that one full runner rotation is equivalent to $3600\Delta t$. Moreover, five internal loops have been selected for each timestep. Every solution has been set to last for ten runner rotations, where the solutions convergence criterion was set as the residuals Root-Mean Square below 10^{-5} . For the sake of checking the validity of numerical results obtained using the above presented numerical simulation scheme, simulation results in terms of the machine external performance characteristics viz developed head (H), efficiency (η), and power (P) have been

compared to experimental ones as shown in Fig.7. To do so, numerically found variations of each the above three parameter have been averaged to come up with final parametric values. The error between the numerical and experimental data is globally found to be less than 5%, which in other words means that numerical results obtained using this numerical scheme can be reasonably compare to real-life situation. Note that utilized formulas for Head, Power, and Efficiency are shown in Equation 6, 7, and 8, where P_2 , P_1 , ρ , g , n , T , and Q stand for Outlet pressure (Pa), Inlet pressure (Pa), Gravitational acceleration (m/s^2), Runner rotational speed (rpm), Torque (Nm), and Flow discharge (m^3/s).

3. Results and discussion

3.1. Flow field characteristics

In order to investigate the eventual changes in terms of water flow dynamics with the runner for different flow conditions, five turbo-planes within the runner inter-blade flow channels have been considered, expanding from hub side towards the shroud side as SP1, SP2, SP3, SP4, and SP5. Considering the unitary spanwise distance from the runner hub to the shroud, these planes occupy successive positions as 0.1, 0.3, 0.5, 0.7, and 0.9 respectively. In Fig.8, contours of runner flow velocity distribution and corresponding vorticity measure under different flow conditions have been presented. Gradually increasing the machine influx from deep part-load flow conditions, to BEP flow conditions, the highest flow speed has been recorded at flow zones within the vicinities of the runner shroud while the lowest are situated close to hub. Moreover, velocity contours show that higher velocity flow on individual turbo-planes are recorded both on the runner blade's suction side and the runner outlet zones (ROZ). It is also obvious that high speed flow zones on the blade suction side (BSS) get larger along the spanwise direction towards the shroud, and extends farther in the streamwise direction towards downstream flow zones.

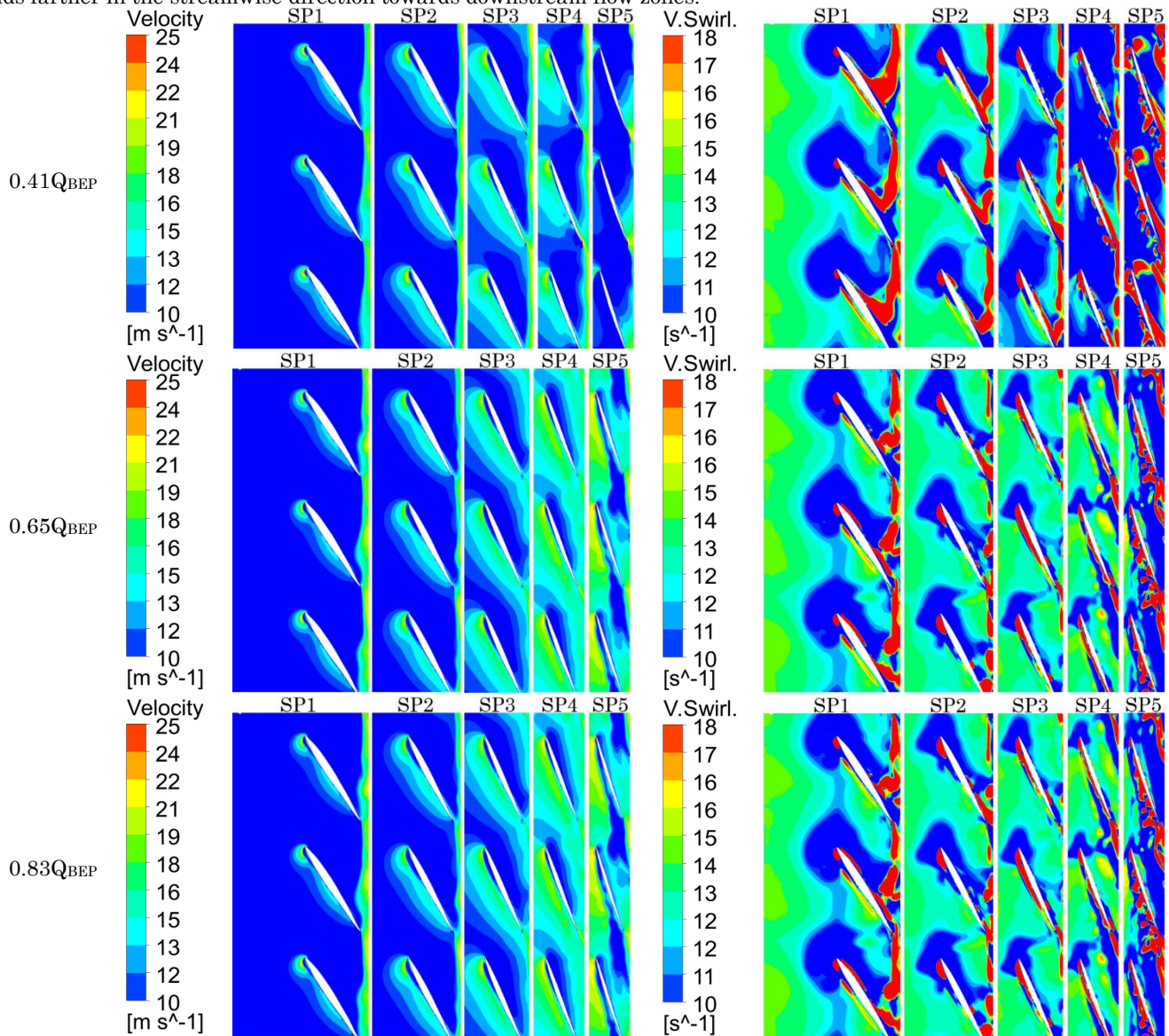


Fig. 8 Contours of flow velocity and vorticity within the runner inter-blade flow zone for different flow conditions

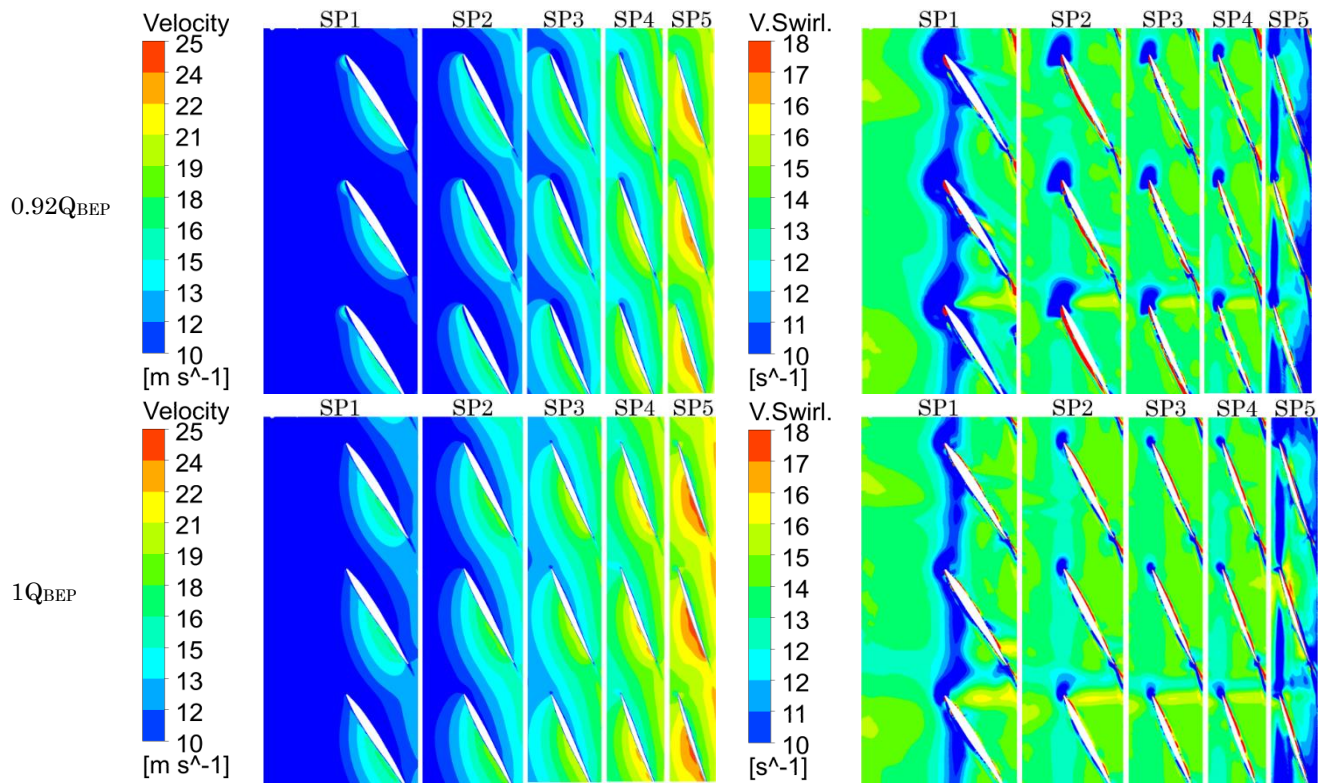


Fig. 8 Continued.

Moreover, the increase in machine influx, has led to a continuous enlargement of flow zones occupied by higher velocity flows on the blade suction side, where they reach the blade trailing edge (BTE) and start to expand towards inter-blade channels' mid-zone, finally touching the blade pressure side's (BPS) trailing zone under upper part-load and BEP flow conditions. This feature is believed to take source from the flow separation that took place both at the runner blade's leading edge and the blade suction side vicinities in downstream flow field. For deep part-load flow conditions, flow separation at blade suction side in downstream zones is amplified by inlet swirling flow, leading to its obvious detachment from the blade leading edge one in the vicinities of the runner hub. This detachment is however found to weaken in the spanwise direction leading a reattachment of both flow separation zones to form one huge zone expanding the whole blade length at flow zones in the vicinities of the runner shroud. Flow separation at the blade leading edge is also found to be linked with large flow incidence angle under deep part-load conditions. It has been found that, with the decrease of the machine flow discharge, the flow incidence angle at the runner blade's leading edge gradually increases, eventually resulting into the formation of huge flow wakes at the same zone, especially within both the runner blade's hub and shroud vicinities under low flow conditions.

In a corresponding way, the representation of flow vorticity measure within the runner inter-blade zone using the velocity swirling strength parameter, not only has it confirmed the above discussed aspects, but also highlighted the presence of two more high vorticity zones. One attached to the runner blade's pressure side at the runner mid-length zone, the other at the blade trailing edge. The later extends opposite to runner rotational direction, swiping the blade trailing edge zone in form of wake structure and obviously occupying the above mentioned high speed flow zone within the runner outlet vicinities, which understandably makes connections with the vaneless space vortex flow, before meeting the next blade's pressure side vortex. It is also found that this high vorticity zone at the runner outlet gradually narrows down in the spanwise direction until the mid-span zone, from where it enlarges again towards the shroud vicinities. At this zone, blade trailing edge-attached wake-like vortices no longer extend enough to reach the next blade's pressure side-attached vortex. Instead, the screw-like vortex emerges at the runner suction side, extending from the blade leading edge's shroud tip region, all the way to downstream zone, where it gradually enlarges towards the runner outlet zone. This structure is attached to the runner shroud, and as any other discussed secondary flows, decays with the increase in machine flow discharge.

To shed more light on the above discussed aspects, through Fig. 9, a visual has been provided on the local pressure distribution mode on the runner blade's suction and pressure surfaces. Starting by the blade suction surface, local pressure distribution mode can be distinctly classified into two zones, namely the low pressure zone at the runner blade's leading edge and vicinities and the high pressure zone at the blade's trailing edge and vicinities. However, there is a smooth transition from one zone to another from the blade leading edge towards downstream, and each zone size varies depending on the considered operating conditions. In good agreement with the above discussed velocity distribution mode, low pressure zone at the blade leading edge spans the whole spanwise blade length from hub to shroud and extends downstream to some blade mid-length distance. In addition this zone extends farther in the shroud region than the hub region. This agrees well with the flow velocity distribution mode on blade suction side as discussed in the above sections, where it was shown that high velocity zones are narrow in hub zone but grow towards downstream in layers above, such as the flow zones close or within the shroud vicinities. These low pressure zones close to the shroud are also found to grow farther in the downstream direction with the increase in the machine flow discharge. The above discussed detachment of high velocity zones on blade suction side from the influence of the inlet swirling flow in hub zone, is even obvious in Fig. 9, where a correspondent high pressure sector within the hub vicinities is noticed to grow in size with the machine flow discharge. On the other hand, the situation on blade pressure surface is the opposite to that of suction surface.

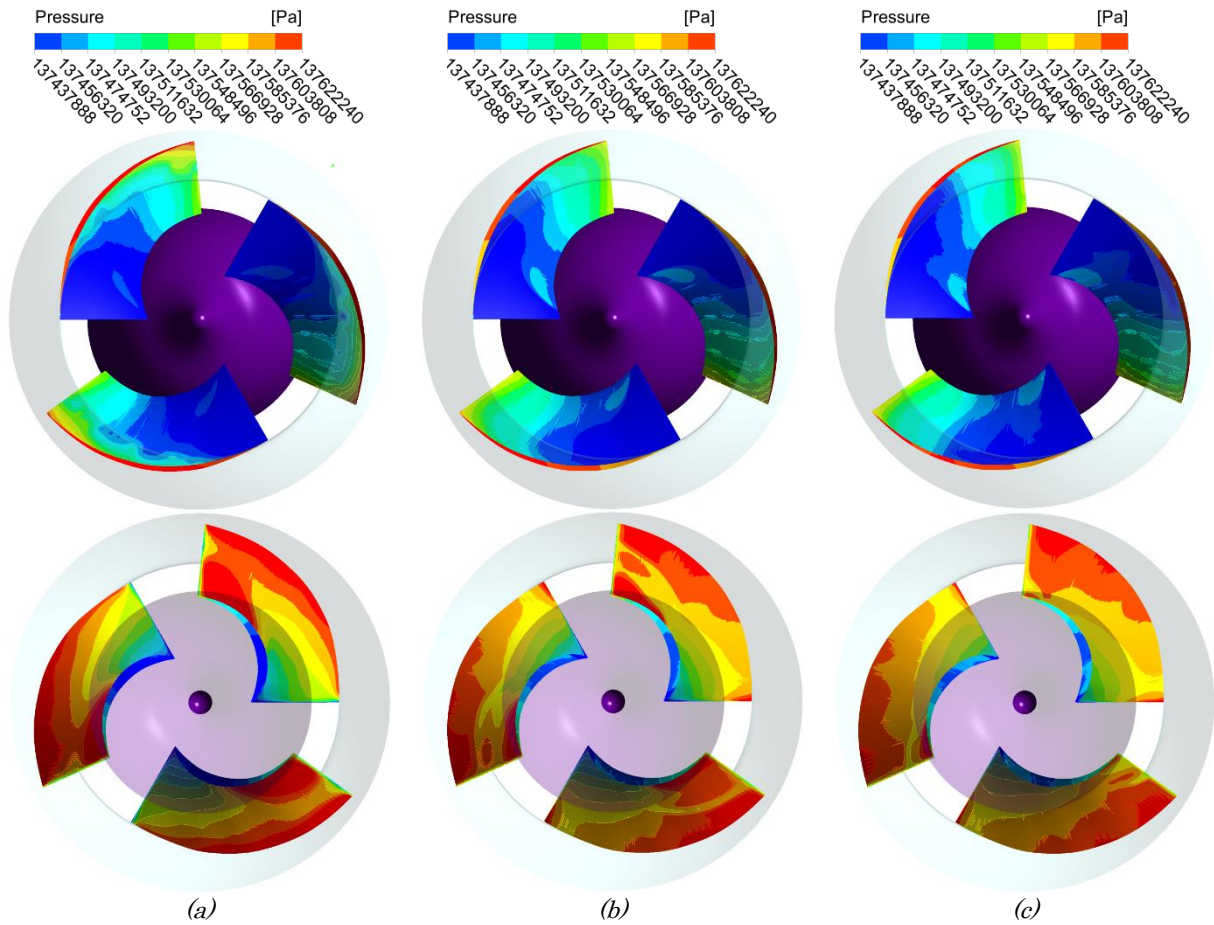


Fig.9 Contours of static pressure on the runner blade's suction (up) and pressure (down) sides under different flow conditions (a) $0.41Q_{BEP}$ (b) $0.65Q_{BEP}$ (c) $0.83Q_{BEP}$.

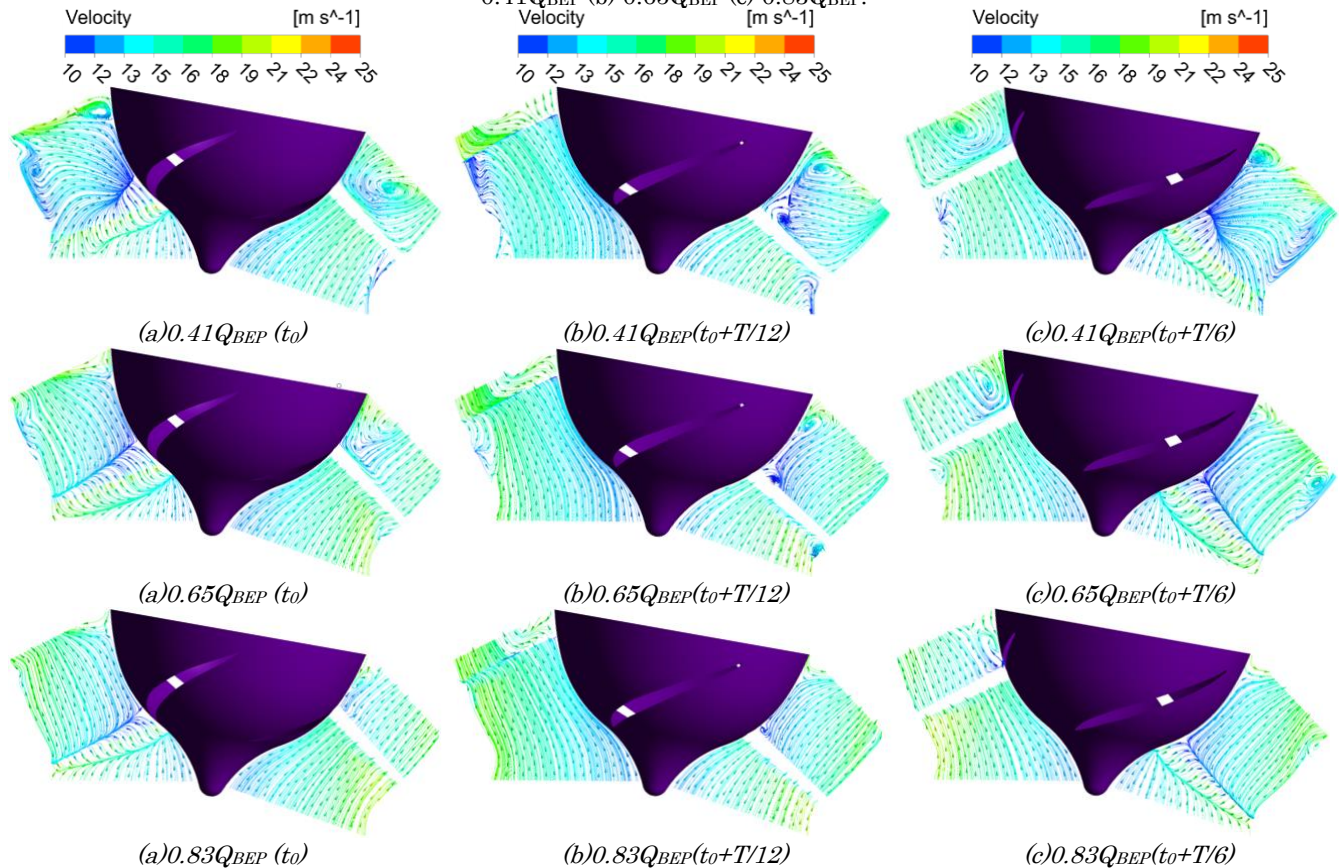


Fig.10 Velocity-colored streamlines within the runner for different flow conditions at different instants.

On this side, high pressure zone are localized at the blade trailing edge vicinal zone and grows big from hub towards the shroud zone. Moreover, the extent of high pressure zone in the shroud vicinities is found to decrease with the increase in machine flow

discharge. This goes in good agreement with the above discussed flow vorticity distribution mode and associated evolution at the same zones.

In Fig.10, velocity-colored flow streamlines on a transversal plane from the runner inlet to the outlet are presented for three successive instants, namely t_0 , $t_0+T/12$, and $t_0+T/6$ respectively, where t_0 stands for initial time and T the runner rotational period. Considering three part-load operating conditions, this figure shows the occurred instant changes in runner flow structures. It is globally shown that runner flow vorticity gets larger and tougher with the decrease of machine flow discharge. Moreover, main flow vortical structures positions seem to remain unchanged despite the change in machine flow conditions. They rather grow or decrease in sizes. In this figure, one would notice both the BPS and BSS-attached vortices and their extensions in shroud vicinities away from the blade, as well as the runner outlet vortex which is mostly within the vicinal zones of the runner hub. The later is found to get larger and expand to mid-span zone and beyond for deep part-load conditions.

Globally speaking, and inline with details as presented in Fig.11, the runner flow vorticity can be classified into four groups respective to their positions, namely the BLE vortex, BSS vortex, BPS vortex, and ROZ vortex. Fig. 11 gives a view of main vortex flow locations in the runner flow zone for the three investigated part-load conditions. Therefore, in this figure, the above mentioned positions have been marked by letters as A, B, C, and D respectively. While the left part of the figure presents the runner assembly with main flow vortex positions indicated, the middle and right sides of the figure show the runner BSS and BPS with the corresponding flow vortices. In total agreement with the above sections, the BLE vortex (A) grows bigger in spanwise direction finally reaching the runner shroud region, from where the BSS vortex (B) starts and extend the whole blade length and beyond, but mainly staying within the shroud vicinal zone. Depending on the considered machine flow conditions, this structure may link up with ROZ vortex (D) at the runner outlet and extend towards the opposite direction of the runner rotation to eventually connect with the precedent blade's BPS vortex (C). This is for instance the case for deep part-load condition as presented in Fig. 11 (a). While all these vortices are found to weaken and eventually vanish as the machine flow conditions move from part-load to rated operating conditions, the BPS vortex does the same while changing locations with different flow conditions. Under deep part-load conditions ($0.41Q_{BEP}$), the BPS vortex extends from blade mid-length to downstream. As the flow increases to part-load conditions ($0.65Q_{BEP}$), it moves spanwise downward to mid-span zone and now extends from BLE to the runner outlet in a string-like form.

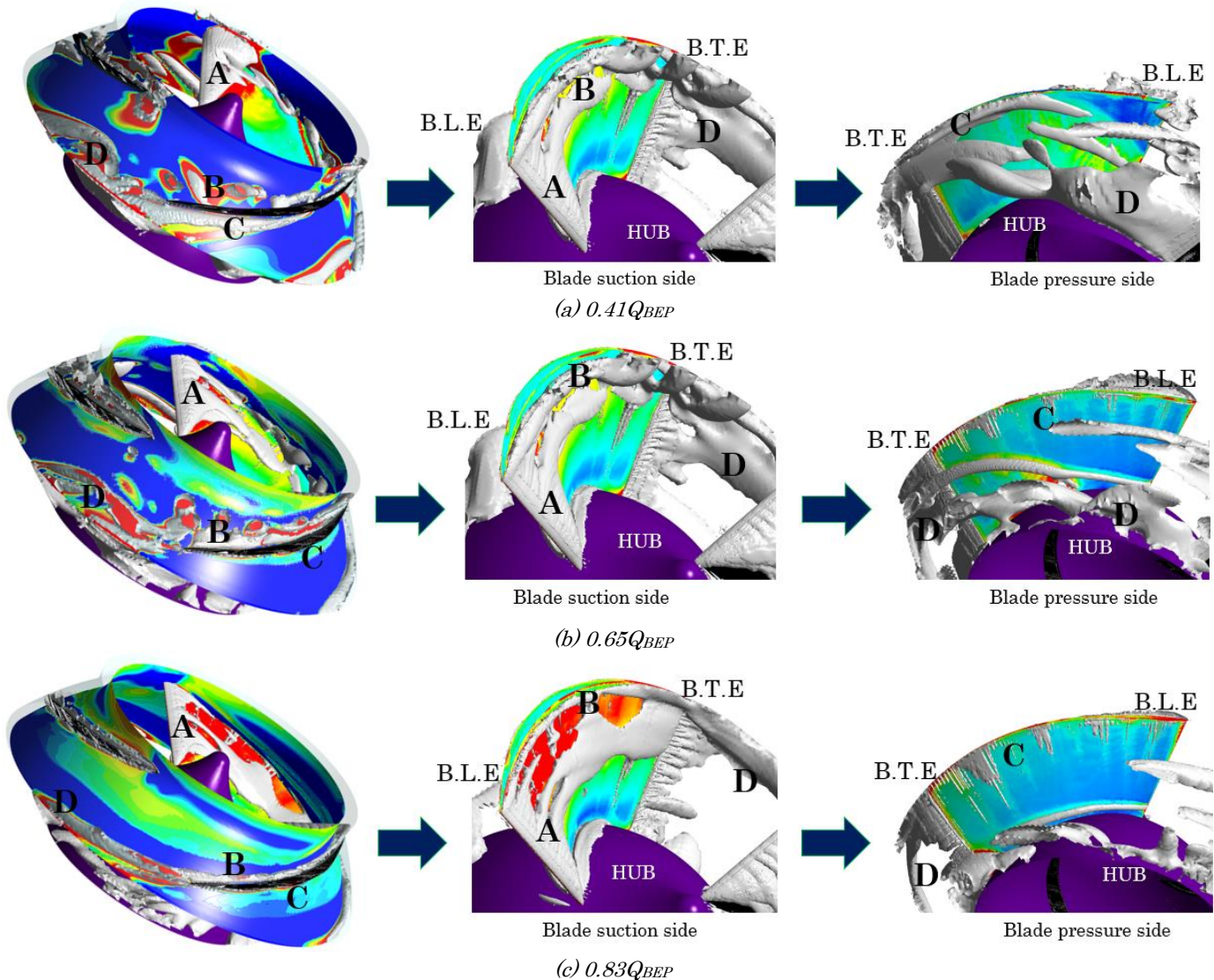


Fig.11 Main flow vorticity zones within the runner for different flow conditions

Under upper part-load conditions it again move downwards to the runner hub's vicinal zone and keep the string-like form extending from the BLE to the ROZ. For the sake of analyzing the flow dynamics within inter-guide vane (GV) flow channels, the

next four figures (Fig. 12 to Fig.14) give a visual of both velocity contours and corresponding velocity-colored streamlines within the diffuser flow zone for different operating conditions. To do so, three consecutive planes, SG1, SG2, and SG3 have been selected in the streamwise direction at positions 1.15, 1.55, and 1.95, considering positions at the diffuser inlet and the diffuser outlet as 1 and 2 respectively. Under $0.41Q_{BEP}$ operating conditions, SG1 being the closest to the vaneless space between the runner and diffuser, exhibits a huge zone of very low flow velocity expanding from the hub to beyond mid-span zone and a narrow high speed flow right at the diffuser's shroud zone.

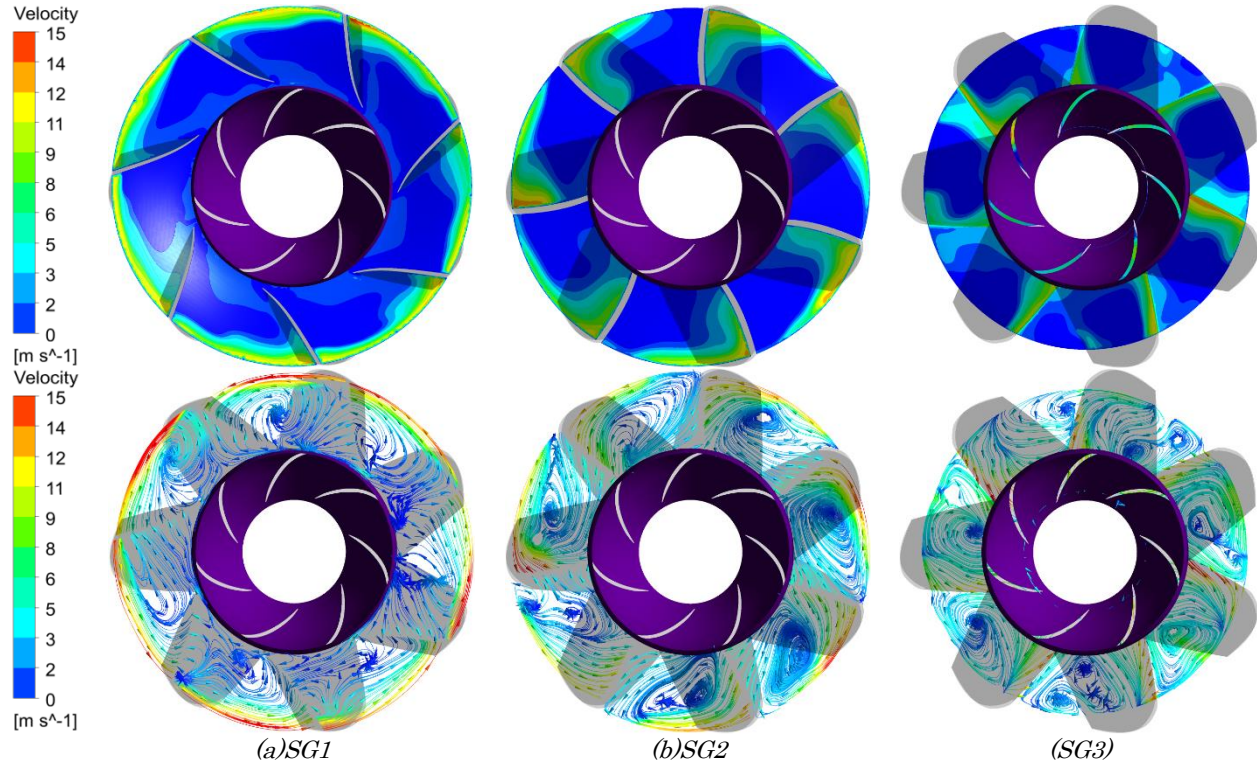


Fig.12 Flow axial velocity contours and streamlines within the diffuser under $0.41Q_{BEP}$ flow conditions for different positions along the streamwise direction.

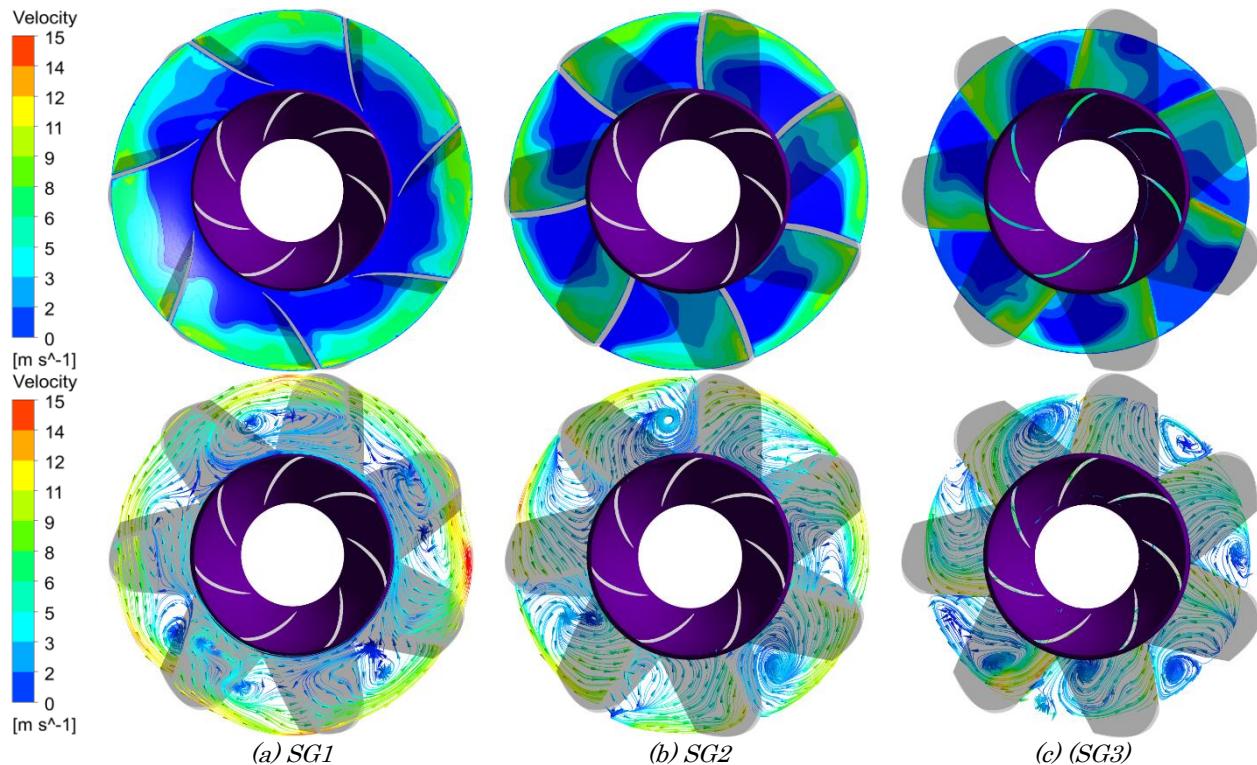


Fig.13 Flow axial velocity contours and streamlines within the diffuser under $0.65Q_{BEP}$ flow conditions for different positions along the streamwise direction.

At this zone, the high speed flow zone is almost circumferentially distributed, where however, in reality, a wider high speed flow zone is attached on the guide vane's suction surface (GSS), extending towards the opposite direction to the runner rotation,

eventually reaching the precedent guide vane's pressure surface (GPS) on its shroud tip region. At this zone, low speed flow occupies almost 80% of the whole flow area. Under the same operating conditions, at the diffuser mid-distance in the streamwise direction (SG2 plane), high speed flow zone on the GSS detaches from the precedent GV and expand spanwise downwards, pushing the low speed flow zone towards the precedent GV's pressure side. At this zone, low speed flow occupies around 75% of the whole flow area. At the diffuser outlet (SG3 plane), low speed zone moves to the center of the channel leaving tiny high speed zones attached to both vanes' pressure and suction surfaces. At this zone, low speed flow zone occupies more than 80% of the available flow area.

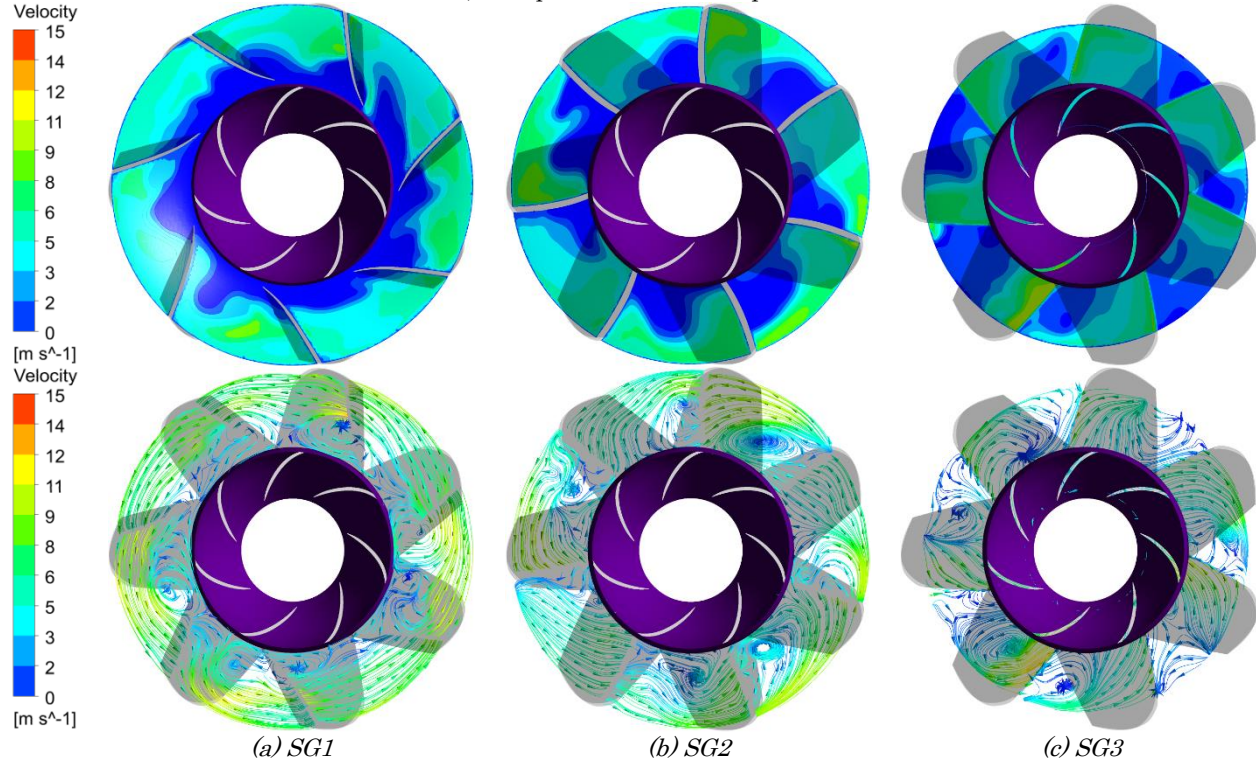


Fig.14 Flow axial velocity contours and streamlines within the diffuser under $0.83Q_{BEP}$ flow conditions for different positions along the streamwise direction.

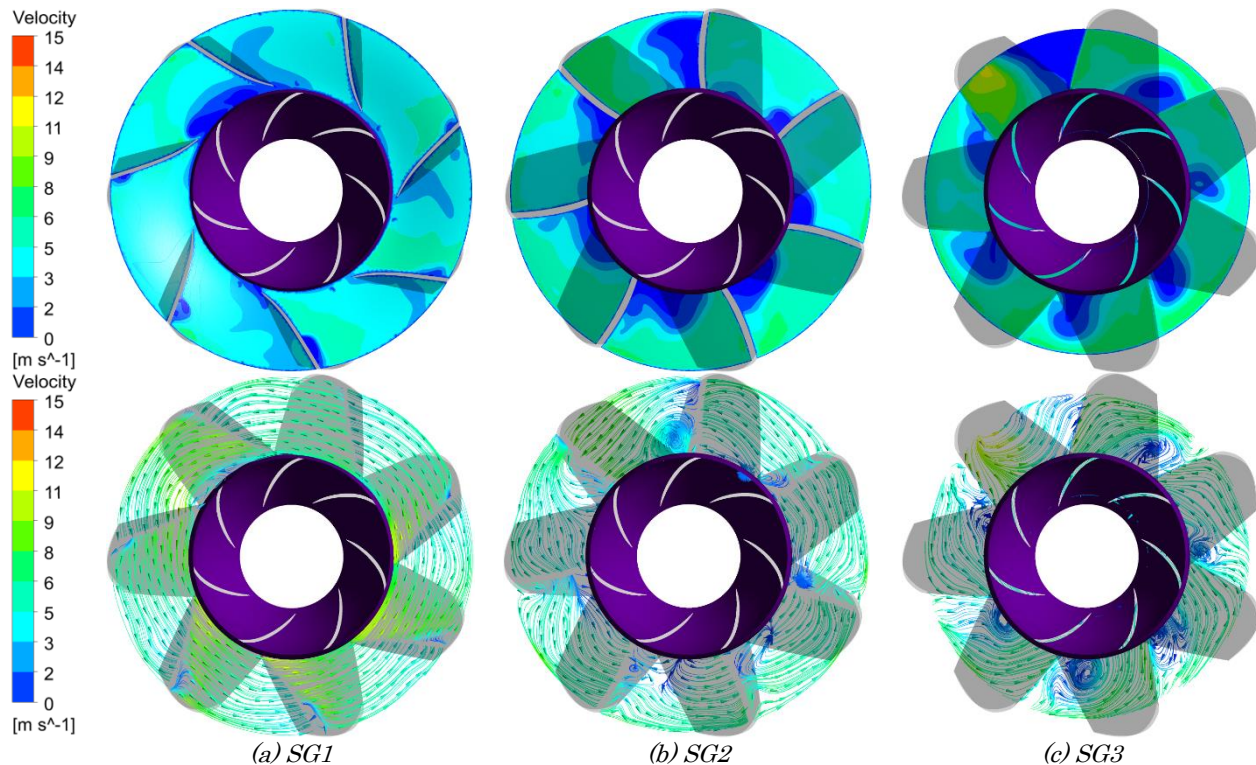


Fig.15 Flow axial velocity contours and streamlines within the diffuser under $0.92Q_{BEP}$ flow conditions for different positions along the streamwise direction.

As shown by correspondent velocity-colored flow streamlines, low flow speed zones are marked by large flow vortices and subsequent backflows. At the diffuser inlet, large vortex and backflow distributes in all channels circumferentially and blocks all

hub flow zones, thus pushing the water to tiny left out flow areas at the shroud, resulting in high speed flows at the shroud and vicinal flow zones. At the diffuser mid-zone, flow vortices occupy a bigger portion from the central flow area to one bounding blade's pressure side, thus leaving the water flowing area at another blade's suction side and a small portion in the shroud vicinal zone. At the outlet, the immergence of a double vortex leads to the occupation of the channel's central zone leaving tiny flowing areas attached to both bounding vanes' surfaces. As shown in the rest of figures (Fig.13 to Fig.15), the flow blockage by the emerged flow vortices and backflows close to the diffuser hub and vicinal flow zones at the diffuser inlet has weakened with the increase in machine flow discharge. Correspondingly, flow vorticity at other flow zone within the diffuser has gradually weakened leading to almost no flow blockage at the diffuser inlet under rated conditions (See Fig.15).

3.2. Pressure field characteristics

In order to investigate the eventual pressure pulsation characteristics as associated with the above discussed flow dynamics, different pressure monitors have been positioned at various locations along the PAT flow full flow passage. Fig.16 gives a visual about the placement of the mentioned pressure monitoring points within the runner flow zone. The specifically investigated zones are the inter-blade flow zones (IB), the runner blade's suction and pressure sides (BS and BP), and the vaneless space (VS) between the runner blades and the distributor's guide vanes. In more details, 9 monitors within the IB zone were positioned at three successive locations in the streamwise direction on the runner's mid-span plane, as shown in Fig.16 (a). Moreover, three spanwise layers have been considered when positioning pressure monitors on the blade's both surfaces. These layers were located at positions 0.1, 0.5, and 0.9 considering the positions of runner hub and shroud as 0 and 1 respectively, where six monitors have considered on each layer, making it 36 monitors on both the BSS and BPS as shown in Fig. 16 (c). Finally, 24 monitors have been equidistantly positioned on the mid-span plane within the vaneless space (Fig. 16 (b)).

Frequency domain pressure pulsation spectra within the vaneless space for the five investigated operating conditions have been shown in Fig. 17. However, for clarity, and owing to the fact that pressure pulsation characteristics within a specific flow zone under specific operating conditions, are more likely to present little to no differences, only the first five VS monitors have been presented. The vaneless space pressure pulsation characteristics are found to considerably vary with the changing machine operating conditions, where for instance, the pressure pulsation amplitude of the dominant frequencies have continuously increased as the machine flow conditions decreased from rated conditions to part-load, before dropping to almost a half under deep part-load conditions.

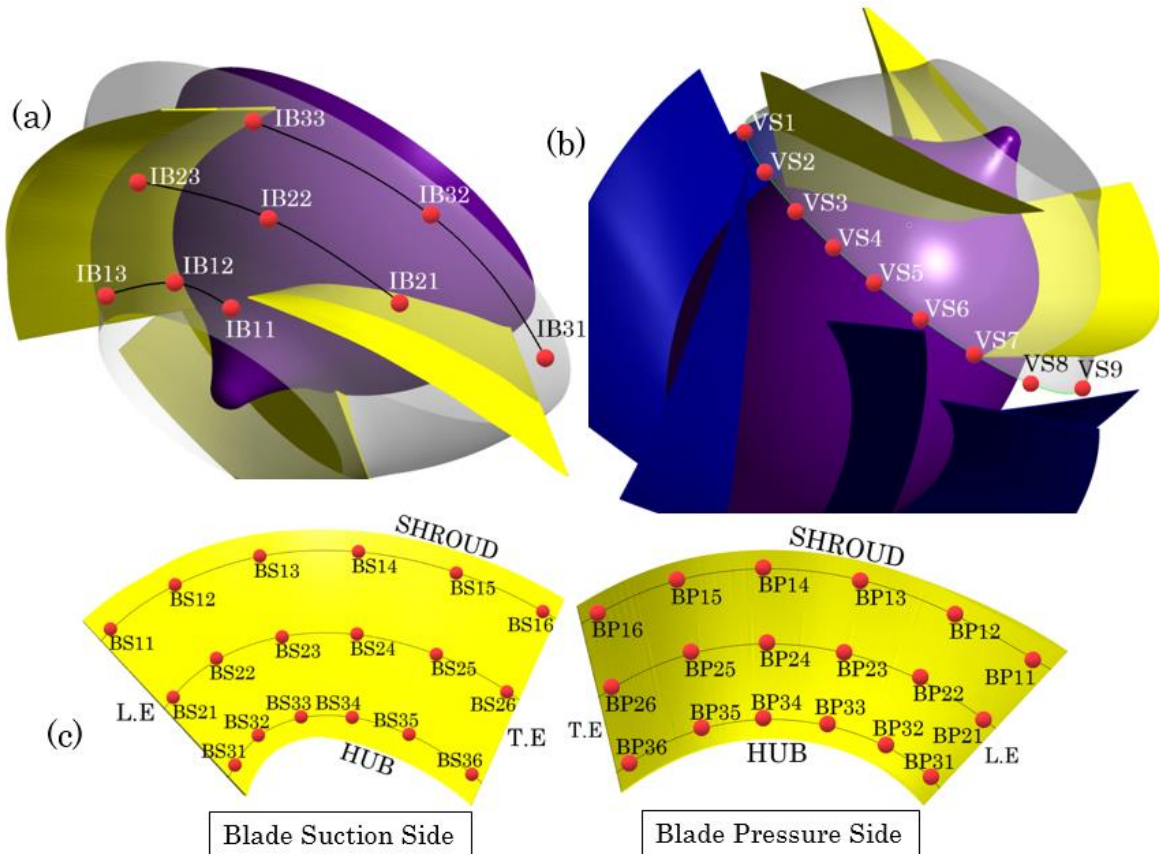


Fig.16 Pressure monitoring locations within the runner flow zone (a) Inter-blade mid-span flow zone (b) Vaneless space flow zone and (c) Runner blade's suction and pressure sides

This aspect agrees well the above discussed flow field characteristics evolution where flow structures got more chaotic as the machine influx decreased. Under rated conditions, associated pressure pulsation spectrum is mainly composed of three frequency components namely the runner rotational frequency (f_n), the dominant frequency $f_i=7f_n$, and its double ($2f_i$). Note that the runner rotational speed is 150rpm, which means that $f_n=2.5\text{Hz}$, while the blade passing frequency (BPF) should be 7.5Hz as the utilized runner has three blades. Moreover, while the BPF stands for perturbations of the distributor flow field by the continuously passing

runner blades, the guide vane passing frequency (GPF) represents the effect of stator guide vanes on rotor flow field. Both BPF and GPF represent the mutual interactions between the runner and distributor; which is widely known as the “Rotor-Stator Interactions” or RSI. This phenomenon is widely recognized as the mighty trigger of large pressure pulsations within hydraulic machinery, which under serious cases may result in the emergence of severe structural vibrations and subsequent machine components breakage or cracks formation. Taking from the grounds that the investigated PAT’s distributor is composed of 7 guide vanes, the above mentioned dominant frequency component f_i is therefore of GPF type. When the machine flow conditions shifted from rated to $0.92Q_{BEP}$ conditions, the amplitude of the dominant frequency component f_i decreases to almost 80% while two new low frequency components (LFCs) emerged, namely $0.25f_i$ and $0.1f_i$, while fn component vanished.

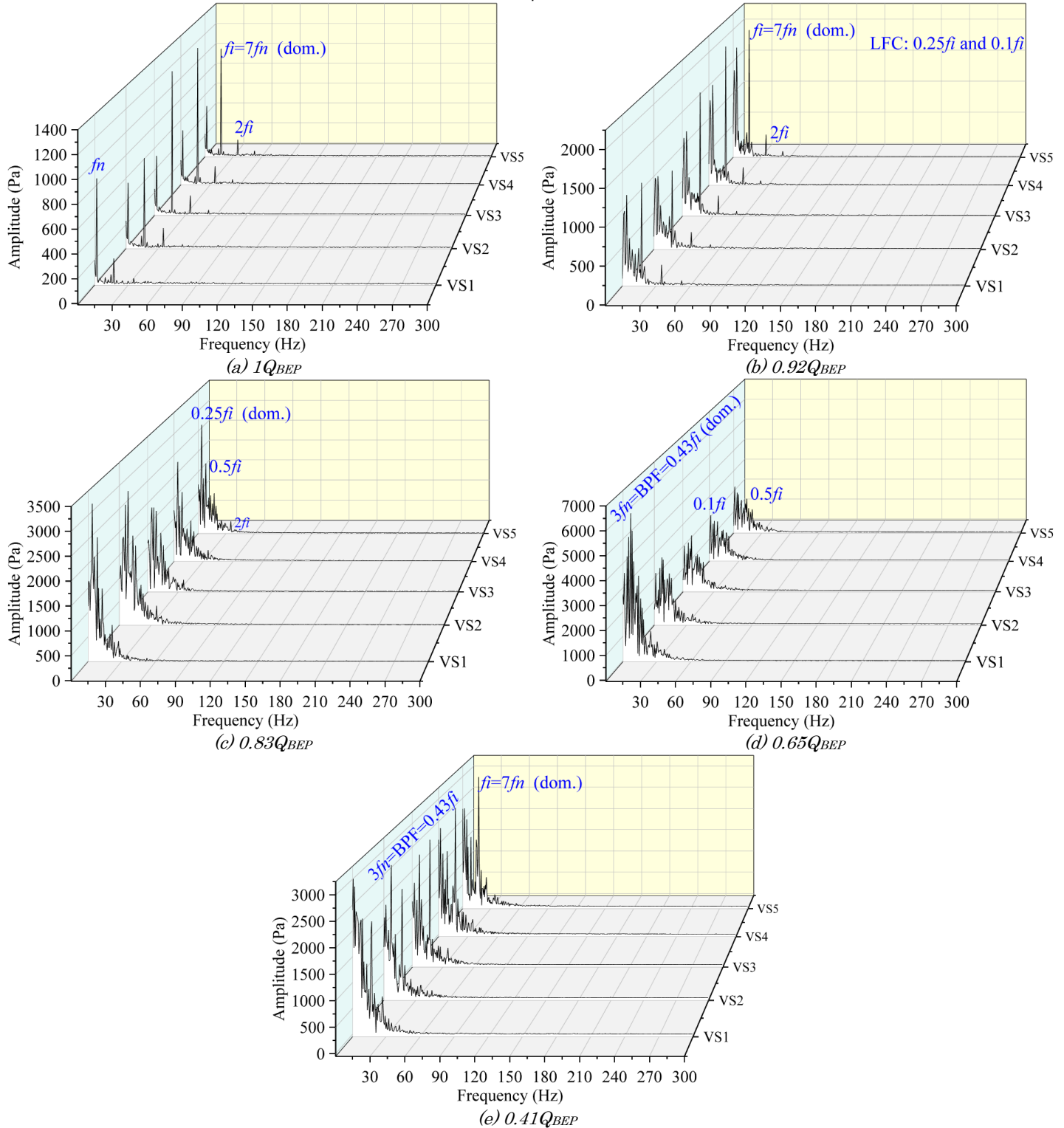


Fig.17 Frequency domain spectra of VS pressure pulsation characteristics under different flow conditions

The emergence of LFC's is connected to the appearance of more secondary flows as the machine operating conditions get away from its nominal operating conditions. Under $0.83Q_{BEP}$ conditions, one of the above mentioned LFCs ($0.25f_i$) became the dominant frequency, where more components in the range from $0.5f_i$ to $2f_i$ surfaced. The $0.65Q_{BEP}$ condition presented the highest of pulsation

amplitudes that was exhibited by the newly emerged blade passing frequency (BPF or $3fn$) component, before falling back to almost its half value under deep part-load conditions where the formerly dominating component f_i retook the lead.

Generally speaking, pressure pulsation spectra have got loaded with more low amplitude frequency components (LAFs) as the machine flow conditions got farther from the optimum conditions, where the last three operating conditions stand out for this aspect. As already mentioned, the emergence of these frequencies and the continuous increase in global pressure pulsation amplitudes are a result of the eventual worsening of flow unsteadiness within the machine flow zones. For the last two operating conditions in particular, associated pressure pulsation spectra are composed of both the local flow unsteadiness-born components and the RSI-born components (BPF and GPF component).

Trying to comparatively get a picture on how machine operating conditions affect the pressure pulsation levels of specific flow zones, a relative pressure pulsation coefficient C_p has been utilized and its expression is written as in Equation 9, where n , $P_{i\max}$ and $P_{i\min}$ stand for number monitors in the same location, individual point's maximum and minimum pressure pulsations amplitudes respectively.

$$C_p = \frac{1}{n\rho gH} \sum_{i=1}^n (P_{i\max} - P_{i\min}) \quad (9)$$

In this respect, Fig.18 presents the pressure pulsation amplitudes distribution mode at different locations within the runner under the five investigated operating conditions. Selected zones are the runner inter-blade channel, vaneless space between the runner and the distributor, as well as the runner blade pressure and suction surfaces. Regardless of the considered operating conditions, pressure pulsation amplitudes within the inter-blade flow channels are generally found to increase in the streamwise direction from the runner inlet to the outlet. Moreover, as again shown in Fig.18 (a), pressure pulsation amplitudes within the inter-blade flow channels are found to globally increase with the decreasing machine flow discharge, leading to the rated conditions presenting the least of pressure pulsation levels while deep part-load conditions exhibited the highest of pulsation amplitudes. On the other hand, for the vaneless space zone (Fig.18 (b)); having generally exhibited the highest level of pulsation amplitudes as compared to other runner flow zones, part-load operating conditions recorded the highest amplitudes, while the least was still with the rated conditions.

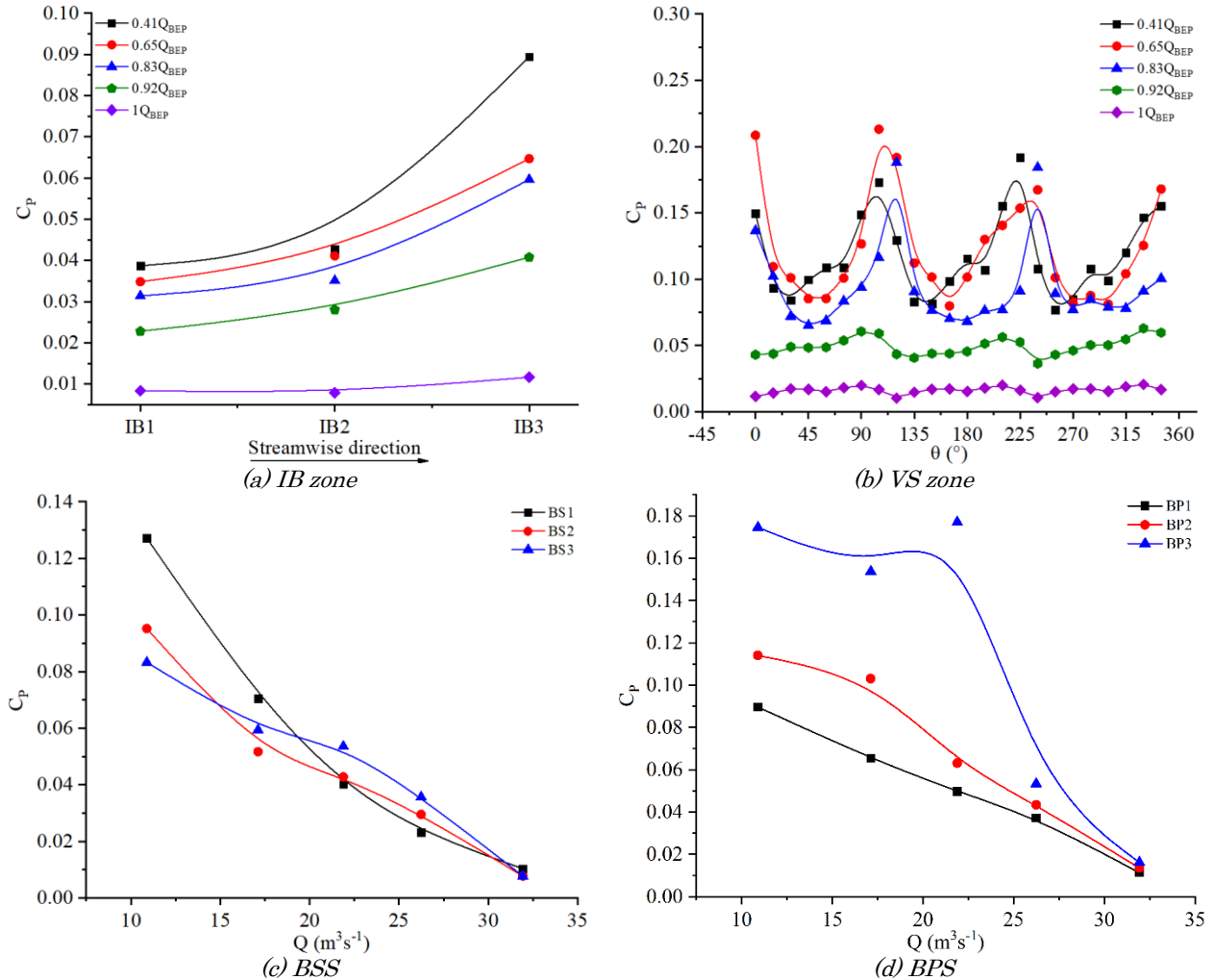


Fig.18 Pressure pulsation amplitude variation within the runner and vaneless space flow zone zones

This is to say that, as also presented through Fig.17, vaneless space pressure pulsation amplitudes increased with the decrease in machine flow discharge, reaching its peak under part-load conditions (0.65 Q_{BEP}) before dropping again towards deep part-load conditions. The VS pressure pulsation amplitudes distribution is more symmetric under both rated and upper part-load conditions (1 Q_{BEP} and 0.92 Q_{BEP}), whereas it gets obviously asymmetric for the last three operating conditions. This somehow aligns well the

emergence of many LAFCs in the VS pressure pulsation frequency spectra, where they were believed to be linked to the worsening of flow unsteadiness under the last three conditions.

Pressure pulsation variation mode on runner blade's suction and pressure sides is presented in Fig.18 (c) and (d) respectively. While BS1 (BP1 on pressure side) is the closest layer to the runner shroud, BS3 (BP3 on pressure side) is the closest layer to the hub. Pulsation amplitudes on blades' both surfaces are generally found to continuously increase with the decrease in machine flow discharge. The highest of pressure pulsation amplitudes on the BSS has been recorded in the runner shroud zone and vicinities under deep part-load conditions, and it decreased in the spanwise direction towards the runner hub region. The randomness in pulsation amplitudes distribution on BSS for other flow conditions is linked with the noticed high flow activity at the same zone. On the other hand, the BPS's lowest pressure pulsation amplitudes were recorded within the runner shroud zone and vicinities under deep part-load conditions, and increased in the spanwise direction downwards to the hub zone. Unlike the BSS case, this trend has been kept for the entire range of investigated flow conditions. Pressure pulsation distribution mode on blade's both surfaces depends on frequent positions of flow vortical structures within this area. As also shown through Fig. 11, flow vortices formation and activity has been seen at the BSS's shroud zone, which was classified as type B. On the other hand, type C on BPS's shroud zone has been observed only under deep part-load condition and vanished for other flow conditions. This somehow correlates with the above discussed fact that BSS recorded the highest pulsation amplitudes in shroud vicinities while it was the hub zone for BPS.

3.3. Effect of runner blade angle

Having investigated the PAT flow structures formation mechanism and associated pressure pulsation characteristics and distribution modes for different operating conditions, the present study has also tried to investigate the effect of runner blade geometric design on the same. In this respect, the runner blade angle γ has been selected as the testing parameter, where its three values, namely $\gamma_1=-2^\circ$, $\gamma_2=0^\circ$, and $\gamma_3=+2^\circ$, were selected for farther analysis. Investigations on incurred changes in PAT flow dynamics with the varying blade angle have been conducted where only two flow conditions, namely the deep-part-load and rated conditions, have been studied.

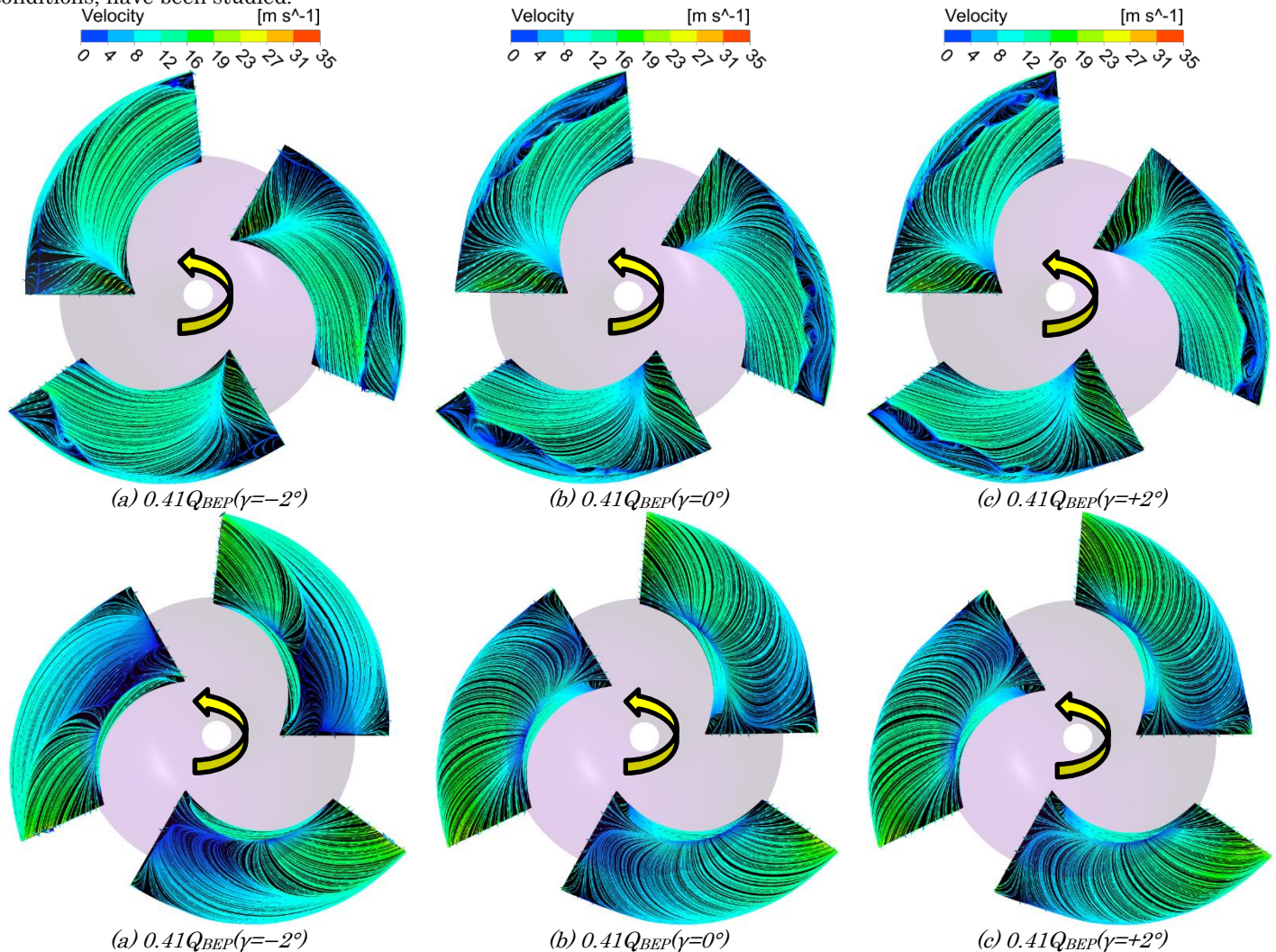


Fig.19 Velocity-colored streamlines on runner blade's suction (up) and pressure surfaces (down) under deep part-load flow conditions ($0.41Q_{BEP}$) for different blade angles.

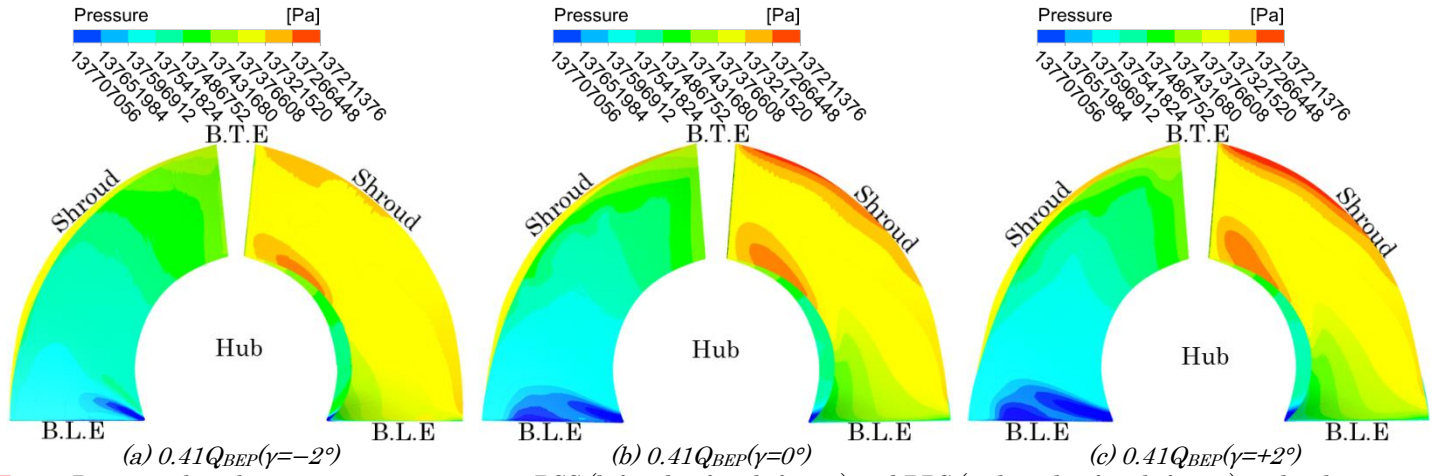


Fig.20 Pressure distribution contours on runner BSS (left side of each figure) and BPS (right side of each figure) under deep part-load conditions ($0.41Q_{BEP}$).

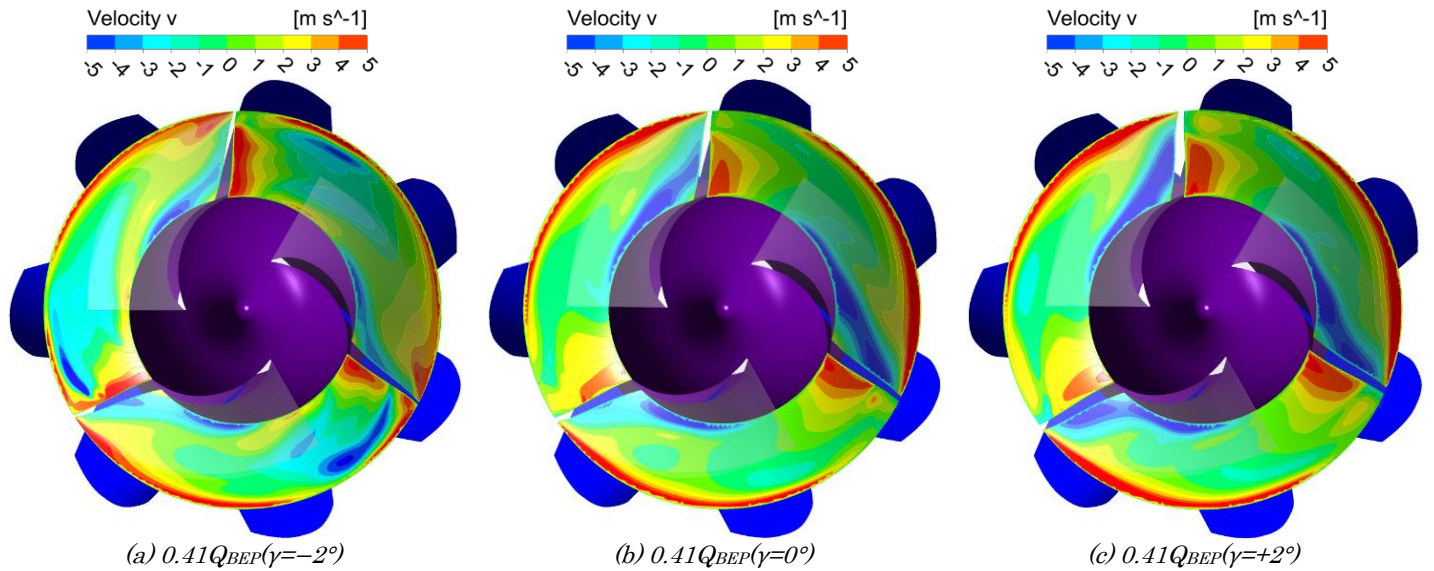


Fig.21 Axial velocity contours at the runner outlet for the three blade angles under deep part-load conditions ($0.41Q_{BEP}$).

Investigations on incurred changes in PAT flow dynamics with the varying blade angle have been conducted where only two flow conditions, namely the deep-part-load ($0.41Q_{BEP}$) and rated conditions ($1 Q_{BEP}$), have been studied.

In Fig.19, velocity-colored streamlines on the blade's suction and pressure surfaces for the three blade angles, namely $\gamma_1=-2^\circ$, $\gamma_2=0^\circ$, and $\gamma_3=2^\circ$, under deep part-load conditions, are shown. It is obvious that the change of runner blade angle has altered the flow structures formation mechanism on both surfaces, where for instance, the formerly mentioned BSS flow vortices at runner shroud and vicinal zones are comparatively weak when using -2° blade angle. These vortices were formerly classified as type B (see Fig.11). However, with the blade angle increase from -2° through 0° to 2° , these vortices eventually got wider occupying more space and got stronger. In addition, the high speed flow zone at the runner blade's leading edge has grown wider with the increase of runner blade angle. Associated flow separation in form of wakes had been classified as type A in Fig.11. On the other hand, the BPS flow field has seen the runner hub-attached vortex flow grow stronger as the blade angle increased. These flow vortices within the hub vicinities are the reason why BPS's hub pressure pulsation amplitudes were the highest as compared to other locations along the spanwise direction as shown in Fig.18 (d). These vortices, partly linking to VS flow unsteadiness within the blade trailing edge vicinal zones, has caused the flow reversal movement back to inter-blade channels, where corresponding hub backflow zone grew wider with the increase in runner blade angle. To support the above discussed truths, Fig.20 and Fig.21 have been used to give more details about the runner flow and pressure field changes taking place as the runner blade angle increases. In Fig.20, pressure distribution mode on both BSS and BPS and eventual changes as the blade angle increased has been presented. In this figure, and in line with the above presented details, BLE low pressure zone on the BSS corresponding to type A flow separation zone has grown wider with the blade angle increase. Moreover, on the BPS, two high pressure zones can be seen both and the shroud and hub zones, corresponding to type C vortex flow and hub backflow zones. Both high pressure zones are seen to increase in size and strength as the blade angle increased. In line with the above, flow axial velocity contours within the vicinal flow zone of the runner outlet have been shown in Fig. 21. At this zone, two high velocity zones can be commonly found at each of the three investigated blade angles. The first is attached on the BSS where extends from hub to over mid-span zone but doesn't reach the shroud side. The second is attached to the shroud and extends from the first blade's pressure side vicinal zone towards the next blade's suction side but doesn't exactly reach. Both zones are globally found to get thicker and stronger with the blade angle increase. In addition, one low axial velocity zone (negative axial velocity or backflow) is commonly found at each of the three investigated blade angles.

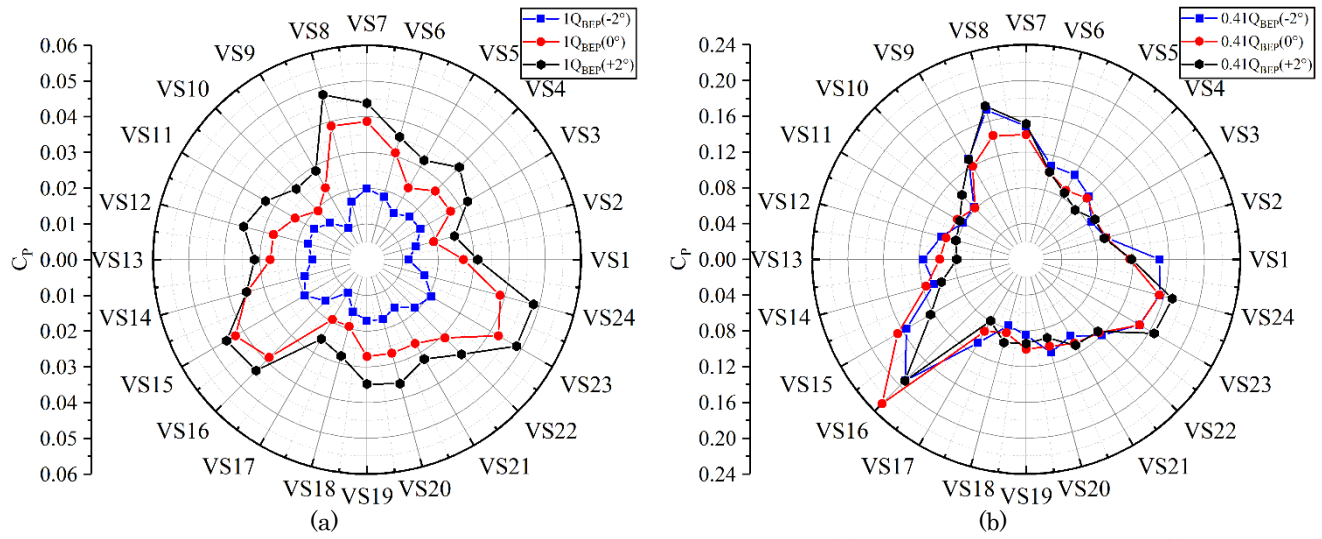


Fig.22 VS pressure pulsation distribution mode for three runner blade angles (a) Rated operating condition (b) Deep part-load operating condition

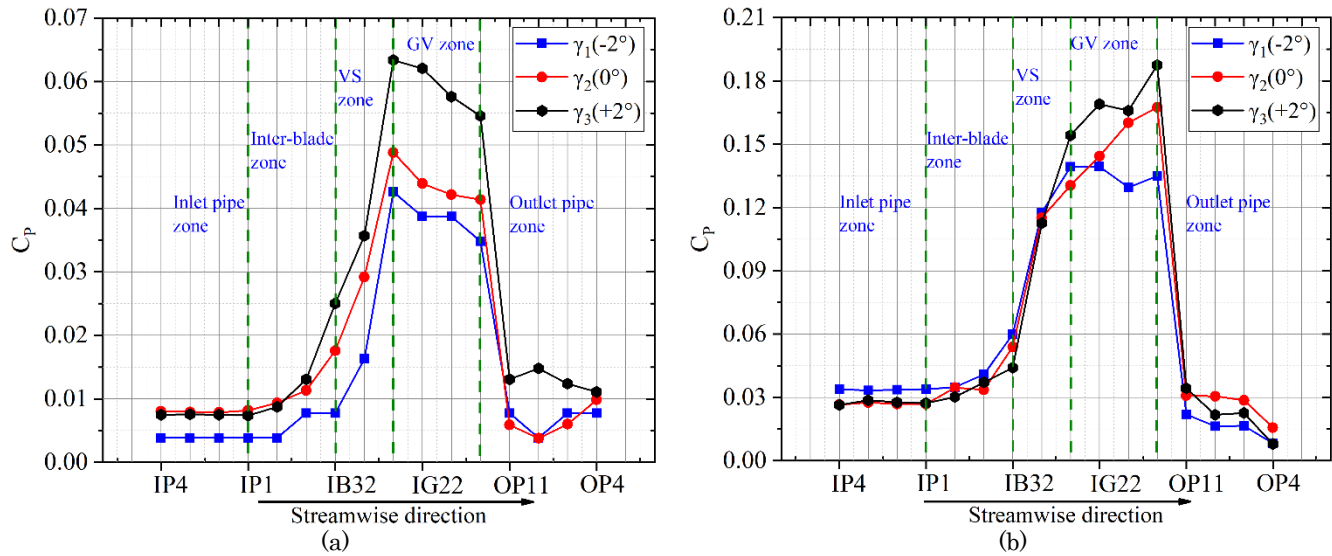


Fig.23 Pressure pulsation distribution in the streamwise direction for different γ values (a) Rated operating condition (b) Deep part-load operating condition

This zone is attached to the hub and is detached from the BPS in the first place, but extends obliquely towards the BPS to finally touch it at the mid-span's vicinal zone. This zone is finally found to grow in size occupying more area both at the hub and on the BPS, where it reaches the mid-span zone under $\gamma=0^\circ$ conditions and goes beyond for $\gamma=\pm 2^\circ$. This agrees well with the above presented details where it has been stated that hub-attached backflow zone grows with the blade angle increase. This also explains the increase in shroud-attached high axial flow zone strengthening with blade angle increase. As the hub-attached backflow zone thickens, the corresponding flow blockage pushes more water flow towards the shroud zone. Because of then existent channel flow obstruction, the shroud-attached positive flow gets more and more accelerated with the increase in hub backflow zone. In the same respect, the noticed negative flow zone in the vicinal zone of the BSS towards the shroud side under $\gamma=0^\circ$ conditions (Fig.21 (a)), has also caused the flow blockage at the same zone which led the emergence of shroud-attached accelerated water flow zone that touches the BSS on its shroud tip region. This zone is however found to disappear with the increase in runner blade angle.

Having described PAT flow dynamics and eventual changes as the blade angle increased, an effort has also been done to explore corresponding modifications of pressure pulsation characteristics and distribution mode at different components of the considered computational domain. In this respect, while considering two operating conditions namely the deep part-load and rated conditions, Fig. 22 displays the pressure pulsation amplitudes distribution mode within the vaneless space flow zone for the three investigated blade angles. The pressure pulsation level of deep part-load conditions is globally found to be almost four times higher than the rated conditions. Having considered 24 monitoring locations equidistantly positioned along the VS circumference on the mid-span plane, pulsation amplitude is found to increase with the blade angle under rated conditions, while the situation is complex for deep part-load conditions, where pulsation amplitudes at different locations almost randomly vary following no common law. Nevertheless the highest of VS pulsation amplitudes under deep part-load condition was recorded under $\gamma=0^\circ$ blade angle at VS16 location. In Fig. 23, streamwise pressure pulsation amplitudes distribution mode from the inlet pipe (IP) through inter-blade and inter-guide vane channels (IB and IG) to the outlet pipe (OP), considering the three blade angles, has been presented. Four axially equidistant monitoring locations expanding from the IP's elbow zone to the runner inlet have been selected as IP4, IP3, IP2, and IP1. Within the inter-blade zone, three monitoring locations, namely IB12, IB22, and IB32, have been selected in the streamwise

direction from runner inlet to the outlet’s vicinal zone on the runner’s mid-span plane. Again, considering distributor’s mid-span plane, four successive locations have been selected in the streamwise direction from inlet to the outlet’s vicinal as IG21, IG22, IG23, and IG24. Finally, starting from the OP’s inlet zone to its elbow zone, four more equidistant monitoring locations have been selected as OP11, OP21, OP31, and OP4. Therefore, adding the VS monitoring location, a total of 16 pressure monitoring locations was chosen all along the flow passage route from the IP’s elbow zone to the OP’s elbow zone. As shown in Fig.23, regardless of the considered blade angle or operating condition, pulsation amplitudes within the IP zone are low and almost constant, followed by a gradual rise through the runner IB channels and vaneless space, reaching the highest level of pulsation amplitudes within the inter-guide vanes channels before falling back to the OP zone. Under rated conditions, the PAT model with $\gamma=+2^\circ$ has recorded the highest of pressure pulsation amplitudes at most of flow zone along the investigated range, followed by $\gamma=0^\circ$ and finally $\gamma=-2^\circ$ presenting the lowest pressure pulsation level. Under deep part-load conditions on the other side, $\gamma=-2^\circ$ dominated the whole flow passage from IP elbow through runner IB zone to the VS zone, mostly followed by $\gamma=0^\circ$. Within the distributor’s IG channels, $\gamma=+2^\circ$ retook the lead, again followed by $\gamma=0^\circ$ for the majority of locations. As for the OP flow zone, $\gamma=0^\circ$ dominated, followed by $\gamma=+2^\circ$ and $\gamma=-2^\circ$ respectively. Moreover, Fig. 23(b) gives more clarity to some aspects as presented in Fig.22 (b). Under deep part-load conditions, the averaged values of recorded pulsation amplitudes within the VS zone have shown that, the highest of pulsation amplitudes under these conditions have been recorded with the $\gamma=-2^\circ$ followed by $\gamma=0^\circ$ and finally $\gamma=+2^\circ$. This therefore represents an opposite situation as compared to the rated conditions situation. However, on a global scale, regardless of the machine operating conditions, the distributor inter-guide vanes flow zone recorded the highest of pressure pulsation amplitudes, where among the three investigated blade angles, the PAT model with $\gamma=+2^\circ$ blade angle exhibited the highest of pressure pulsation levels followed by $\gamma=0^\circ$ and $\gamma=-2^\circ$ respectively.

4. Conclusions

In this study, experimental and numerical simulations have been carried out to investigate the performance characteristics of a mixed flow PAT under pump mode, as well as the associated flow structures formation mechanism for different operating conditions and runner blade angles. Having considered five operating conditions ranging between the optimum and deep part-load conditions, and three blade angles namely -2° , 0° , and $+2^\circ$, the following conclusions have been drawn:

- a) Within the decrease of machine flow from optimum conditions to deep part-load conditions, flow unsteadiness gets gradually serious, where the last three conditions ($0.41Q_{BEP}$, $0.65Q_{BEP}$, and $0.83Q_{BEP}$) presented the most disturbed flow structures. Under these conditions, the most dominated regions by flow vortical structures within the runner are runner BLE, the BSS’s shroud side, the BPS’s hub side, and the ROZ; while for the distributor component, hub zone and GPS have been the ones to experience the largest of flow unsteadiness.
- b) Pressure pulsation amplitudes have been found to increase with the decreasing machine flow discharge, reaching its peak under part-load conditions ($0.65Q_{BEP}$) before falling back to deep part-load conditions. Depending on the considered operating conditions, the VS pressure pulsation spectra have been found to be composed of RSI-born frequency components (GPF and BPF) and LFU-born LAFCs. The RSI-associated components served the dominant frequency components for the majority of investigated operating conditions.
- c) The runner blade angle has caused changes in the machine flow and pressure field characteristics, where for instance, with the increase of runner blade angle from -2° through 0° to 2° , the runner flow unsteadiness has only correspondingly worsened. Pressure pulsation levels within the vaneless space flow zone have been found to increase with the blade angle under optimum conditions while the opposite happened under deep part-load conditions. However, on a global scale, the highest pulsation amplitudes were recorded within the distributor components and they increased with the runner blade angle.

Having studied this PAT model’s pump mode flow dynamics for a wide range of flows and three blade angles, next research focus will turn to investigating the same aspects considering the turbine mode of operation, where an attempt is going to be made to establish a relationship between both operating modes’ flow structures formation mechanisms. Moreover, an optimization scheme will be utilized to improve both modes’ flow field characteristics leading to reasonably higher performances in both modes.

Acknowledgements

This study was supported by the Postdoctoral Research Fund of Jiangsu Province (2020Z152), National Natural Science Foundation of China (52009033; 52006053), Natural Science Foundation of Jiangsu Province (BK20200509; BK20200508), and the Fundamental Research Funds for Central Universities (B210202066). The support of Hohai University, China is also gratefully acknowledged.

Acronyms

PAT	Pump as Turbine	RANS	Reynolds Averaged Navier-Stokes equations
BEP	Best Efficiency Point	SST	Shear Stress Transport turbulence model
RBA	Runner Blade Angle	GGI	General Grid Interface
VS	Vaneless Space	LFC	Low Frequency Component
CFD	Computational Fluid Dynamics	FFT	Fast Fourier Transform
BPF	Blade Passing Frequency	LFU	Local Flow Unsteadiness
GPF	Guide vane Passing Frequency	RSI	Rotor-Stator Interactions
LAFC	Low Amplitude Frequency Component	BLE	Blade Leading Edge
BPS	Blade Pressure Surface	BTE	Blade Trailing Edge
BSS	Blade Suction Surface	GV	Guide Vane
ROZ	Runner Outlet Zone	GSS	Guide vane Suction Surface
IEC	International Electrotechnical Commission	GPS	Guide vane Pressure Surface
IP	Inlet Pipe	OP	Outlet Pipe

Latin & Greek letters

y^+	Dimensional height from the wall	fn	Runner rotational frequency
Q	Discharge	n	Runner rotational speed
H	Head	g	Gravitational acceleration
P	Power	η	Efficiency
P	Static pressure	ρ	Density
T	Torque	γ	Blade angle
Z_R	Number of runner blades		

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