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The Effects of Curved Blade Turbine on the Hydrodynamic Structure of a Stirred Tank

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Abstract

This work is aimed at studying the hydrodynamic structure in a cylindrical stirred vessel equipped with an eight-curved blade turbine. Flow fields were measured by two-dimensional particle image velocimetry (PIV) to evaluate the effect of the curved blade turbine. Velocity field, axial and radial velocity distribution, root mean square (rms) of the velocity fluctuations, vorticity, and turbulent kinetic energy were presented. Therefore, two recirculation loops were formed close to the free surface and in the bottom of the tank. Moreover, the highest value area of the vorticity is localized in the upper region of the tank which follows the same direction of the first circulation loop. The turbulent kinetic energy is maximum at the blade tip following the trailing vortices.

Keywords: hydrodynamics structure, stirred vessel, curved blade turbine, PIV

1. Introduction

Stirrer geometry and design in a mechanically agitated tank have been studied over a wide range of design aiming at improving the agitation efficiency. Several studies which have already been carried out were interested in enhancing the vessel property, experimentally and numerically. In fact, the experimental study includes the effect of removal of baffles, the impeller geometrical effects, the number of blades on energy efficiency, and the impacts of solid concentration and particle size on power consumption [1]. Mixing time and pattern in the agitated vessel was also experimentally investigated at various mixing Reynolds numbers [2]. Three types of impellers generated in different liquid flows as well as their position were used to investigate their influences on the kinetic parameters in a batch cooling crystallizer [3]. In addition, particle image velocimetry technique (PIV) was used to carry out the turbulent flow inside a cylindrical baffled stirred vessel with a set of speed ranging from 100 to 350 rpm [4]. Ben Amira et al. [5] also studied the hydrodynamic structure of the flow generated by eight concave blade turbines. Furthermore, PIV technique is also used to estimate the turbulence energy dissipation rate in a stirred vessel generated by a Rushton turbine [6]. In addition, both experimental and numerical techniques were developed simultaneously. In fact, Cruz-Díaz et al. [7] modeled the operation of the FM01-LC reactor coupled with a continuous stirred tank (CST) in recirculation mode. Li et al. [8] used the large eddy simulation

and the PIV technique to calculate the velocity field generated by a Rushton turbine. In-line high shear mixers (HSMs) with double rows of ultrafine inclined stator teeth were experimentally and numerically investigated under different rotor speeds and flow rates [9]. Furthermore, computational fluid dynamics (CFD) simulations were used to investigate the effects of impeller configuration on fungal physiology and cephalosporin C production by an industrial strain *Acremonium chrysogenum* in a bioreactor equipped with conventional and novel impeller configuration, respectively [10]. Navier-Stokes equation in conjunction with the RNG (renormalization group) of the $k-\epsilon$ turbulent model was used to study the turbulent flow induced by the six flat blade turbines (FBT6), the Rushton turbine (RT6), and the pitched blades turbine (PBT6) in a stirred tanks [11]. Finite volume method was employed to solve the Navier-Stokes equations governing the transport of momentum to compare four different turbulence models used for numerical simulation of the hydrodynamic structure generated by a Rushton turbine in a cylindrical tank [12]. Multiple impellers were used in a stirred vessel to form the micro/nano drug particle in the biopharmaceutical classification system [13].

According to the biography, it is interesting to study the effect of the blade shape in order to improve the hydrodynamic structure. In this paper, we are interested in studying the hydrodynamic structure in a cylindrical stirred vessel equipped by an eight-curved blade turbine.

2. Experimental apparatus

In this study, PIV was used to study the hydrodynamic structure generated with the convex blade turbine in a stirred vessel. For the particle image velocimetry, the flow is illuminated by an Nd-YAG 532 nm green pulsed laser source generated in 2×30 mJ. The acquisition of the two-dimensional image data was taken with a CCD camera with 1600×1200 pixels of resolutions. The results are obtained for 170 images at three different azimuthally planes. The average diameter of the seeding particle was equal to $d_p = 20 \mu\text{m}$ with 0.15 g of concentration. The cylindrical vessel was mounted in a squared vessel to decrease reflection which is filled by water. The height of the water is equal to the tank diameter ($D = 300$ mm). Besides, the turbine diameter is equal to the half of the vessel diameter ($T = D/2$), and it is placed in the middle of the tank. Four equally spaced baffles which are placed 90° far from one another were used. The rotation speed of the turbine is equal to $N = 70$ rpm which is equal to a Reynolds of $Re = 26,250$. The velocity at the tip of the impeller is equal to $U_{tip} = 0.55$ m/s. The velocity speed is settled by using an electrical motor placed at the top of the tank and controlled automatically by the computer. In addition, the azimuthally plane of the investigation is localized at $\theta = 10^\circ$ from the blade. The angular position between two successive blades is settled through a position sensor localized near the electrical motor.

Figure 1 shows the different impellers geometry for the one-staged system namely, flat, concave, and convex blade impeller. **Figure 2** shows the staged system. In fact, the highest impeller is localized at the third position according to the free surface, and the lowest one is localized at the third position according to the bottom. For the first configuration (**Figure 2(a)**), the flat blade impeller is localized at the highest position and the concave blade impeller at the bottom (PD8 h, PI8 concave). For the second configuration, the impeller at the bottom is replaced by the convex impeller (PD8 h, PI8 convex). For the third and the fourth configurations, the flat blade impeller is localized at the bottom, and the highest impeller is occupied by the concave and the convex blade impeller, respectively (PD8 b, PI8 concave; PD8 b, PI8 convex).

Figure 1.
 One-staged system.

Figure 2.
 Staged system.

3. Data analysis

For a given point, the instantaneous velocity components u_{ij} in terms of a mean velocity $\bar{u}_i$ and a temporal fluctuating term u'_{ij} is given as

$$u_{ij} = \bar{u}_i + u'_{ij} \tag{1}$$

where i refers to the velocity component r , z , or θ and j refers to the instant at which the velocity was measured for each acquisition.

The mean velocity is calculated as follows:

$$\bar{u}_i = \frac{1}{N_n} \sum_{j=1}^{N_n} u_{ij} \tag{2}$$

With N_n refers to the snapshot total number. The average of the temporal fluctuating terms which known as a root mean square (rms) value is given by

$$u_{i,rms} = \left(\overline{u'^2_{ij}} \right)^{1/2} = \left(\frac{1}{N_n} \sum_{j=1}^{N_n} (u_{ij} - \bar{u}_i)^2 \right)^{1/2} \tag{3}$$

For two-dimensional velocity data, in the Cartesian coordinate, the turbulent kinetic energy is equal to

$$k = \frac{3}{4} (\overline{u^2} + \overline{v^2}) \quad (4)$$

The proper orthogonal decomposition (POD) is obtained by computing the auto-covariance matrix (R). According to Liné et al. [14], the matrix of instantaneous velocity vector data is calculated as

$$M = \begin{bmatrix} u_1(1) & \dots & u_{Nn}(1) \\ \vdots & & \vdots \\ u_1(n) & \dots & u_{Nn}(n) \\ v_1(1) & \dots & v_{Nn}(1) \\ \vdots & & \vdots \\ v_1(n) & \dots & v_{Nn}(n) \end{bmatrix} \quad (5)$$

Nn refers to the number of snapshots, and n refers to total number of the interrogation area of the whole flow field. The auto-covariance matrix, which is associated to the fluctuating velocity components for each snapshot, is calculated as follows:

$$R = \frac{1}{Nn} M \cdot M^T \quad (6)$$

Eigenvalues are calculated by solving the Fredholm integral eigenvalue problem and Karhunen-Loeve analysis:

$$\iint_{\Omega} R(x, z, x', z') \overrightarrow{\phi^{Ki}}(x', z') dx' dz' = \lambda^{ki} \overrightarrow{\phi^{Ki}}(x, z) \quad (7)$$

where Ki refers to the POD mode and Ω refers to the domain of interest. The eigenfunction is calculated as follows:

$$\phi = \frac{\sum_{n=1}^{Nn} a_n^{Ki} u^n}{\left\| \sum_{n=1}^{Nn} a_n^{Ki} u^n \right\|} \quad (8)$$

4. Experimental results

4.1 Velocity field

Figure 3 shows the velocity field of different types of curved blade turbines for one-staged system. According to these results, two circulation loops were observed, in which the first one is localized in the upper region of the tank near the free surface and the second one is localized in the bottom of the tank. A radial jet is

Figure 3.
Velocity field for one-staged system.

described at the blade tip. In fact, the velocity has some deviation in the other azimuthal planes by the effect of the propagation of trailing vortices. The jet flow is more intensive for the convex blade turbine. For the concave blade, a maximum velocity is created at the top edge that deviated to the blade tip. Afterward, near the wall of the vessel, the velocity is divided into the upward and downward flow. Then, it returns to the shaft to create the circulation loop. As a matter of fact, the concave configuration produces a larger lowest loop than the other configurations. Hence, the region located below the impeller (at the bottom of the tank) is more turbulent than the flat and the convex configurations. Moreover, the distribution of the turbulent flow at the upper and the downer regions is more similar at the convex configuration.

The flow distribution of the flat and the curved blade is similar to that presented by Driss et al. [15] for a laminar flow. In fact, a radial jet is created, and then an axial flow is obtained by the wall effect. In addition, the flow slows down significantly far away from the impeller. The shape and the position of the recirculation loops and the trailing vortices are affected significantly by the blade design [15, 16].

The coordination of the center of the highest loops (z_h^* , r_h^*) and the lowest loops (z_b^* , r_b^*) and its radial and the axial extension (z_b , r_b , z_h , r_h) are presented in **Table 1**.

	Flat impeller		Concave impeller		Convex impeller	
Highest loops	z_h^* (z/H)	0.78	z_h^* (z/H)	0.78	z_h^* (z/H)	0.78
	r_h^* (r/R)	0.76	r_h^* (r/R)	0.81	r_h^* (r/R)	0.76
	z_h (z/H)	0.32	z_h (z/H)	0.21	z_h (z/H)	0.32
	r_h (r/R)	0.42	r_h (r/R)	0.33	r_h (r/R)	0.4
Lowest loops	z_b^* (z/H)	0.32	z_b^* (z/H)	0.25	z_b^* (z/H)	0.27
	r_b^* (r/R)	0.72	r_b^* (r/R)	0.75	r_b^* (r/R)	0.8
	z_b (z/H)	0.22	z_b (z/H)	0.37	z_b (z/H)	0.33
	r_b (r/R)	0.4	r_b (r/R)	0.3	r_b (r/R)	0.3

Table 1.
Loop coordination and shape of the one-staged system.

Figure 4.
Velocity field for staged system.

		PD8 h, PI8 concave	PD8 h, PI8 convex	PD8 b, PI8 concave	PD8 b, PI8 convex
Highest loops	$z_h^* (z/H)$	0.78	$z_h^* (z/H)$ 0.76	$z_h^* (z/H)$ 0.74	$z_h^* (z/H)$ 0.82
	$r_h^* (r/R)$	0.7	$r_h^* (r/R)$ 0.71	$r_h^* (r/R)$ 0.7	$r_h^* (r/R)$ 0.65
	$z_h (z/H)$	0.29	$z_h (z/H)$ 0.23	$z_h (z/H)$ 0.26	$z_h (z/H)$ 0.26
	$r_h (r/R)$	0.46	$r_h (r/R)$ 0.44	$r_h (r/R)$ 0.44	$r_h (r/R)$ 0.46
Lowest loop	$z_b^* (z/H)$	0.21	$z_b^* (z/H)$ 0.25	$z_b^* (z/H)$ 0.19	$z_b^* (z/H)$ 0.26
	$r_b^* (r/R)$	0.74	$r_b^* (r/R)$ 0.67	$r_b^* (r/R)$ 0.75	$r_b^* (r/R)$ 0.7
	$z_b (z/H)$	0.33	$z_b (z/H)$ 0.32	$z_b (z/H)$ 0.24	$z_b (z/H)$ 0.36
	$r_b (r/R)$	0.46	$r_b (r/R)$ 0.46	$r_b (r/R)$ 0.43	$r_b (r/R)$ 0.47

Table 2.
Loop coordination and shape of the staged system.

In addition, the maximum velocity is greater for the convex blade turbine and weaker for the concave blade turbine due to the interaction between the blade and the flow that is lower for the convex blade and greater for the concave blade (flat turbine $U_{\max} = 0.22 U_{\text{tip}}$, concave turbine $U_{\max} = 0.18 U_{\text{tip}}$, convex turbine $U_{\max} = 0.35 U_{\text{tip}}$).

For the staged system (**Figure 4**), the flow becomes more turbulent, and the loops can reach the free surface as the bottom of the tank. In addition, an oblique flow is created between the two impellers. The highest velocity is produced in the case of the second configuration ($U_{\max} = 40\% U_{\text{tip}}$) (a, $U_{\max} = 34.54\% U_{\text{tip}}$; c, $U_{\max} = 32.72\% U_{\text{tip}}$; d, $U_{\max} = 32.72\% U_{\text{tip}}$). The position and the shape of the recirculation loops of each configuration are presented at **Table 2**.

The combination of inclined blade turbine and flat turbine shows no great change in terms of acceleration and shape of the recirculation loops. This found locks similar to that developed by Bereksi et al. [17] for the combination between the Rushton and the curved blade. In addition, they proved that the gas holdup is better by the combination between the curved and the Rushton turbine than by the combination between two Rushton turbines.

4.2 Radial and axial velocity

Figure 5 shows the distribution of the radial velocity component of the curved blade turbine. According to these results, the highest value region is localized at the

Figure 5.
Radial velocity for one-staged system.

end of the turbine. The bulk region of the tank is described with the lowest value. The flat blade turbine generates a larger radial velocity. Hence, the area of the maximum radial velocity is larger than the other configurations. The maximum radial velocity is spread to reach places farther than the blade, which can be explained by the ability of the blade shape to generate trailing vortices. The development of the radial velocity component for the concave blade configuration is closer to the axial turbine. In fact, the maximum velocity area is localized at the top edge of the blade and spreads to the same direction as the Von Karman vortex street. The highest value of the radial velocity component is generated by the convex blade configuration that is equal to $u_{\max} = 0.35 U_{\text{tip}}$, whereas it is equal to $u_{\max} = 0.15 U_{\text{tip}}$ for the other configurations. However, the maximum value remained closer to the blade tip. Therefore, it can be seen that the convex shape of the blade gives the ability to the turbine to move easily within the water and transmit more velocity while not giving it enough capacity to expand much.

For the staged system (**Figure 6**), the maximum value of the radial velocity component is localized between the two blades. This explains the oblique direction of the velocity field at this region. The largest maximum area is defined at the association of the concave and the flat turbines that confirm that the maximum value cannot spread with the convex shape. In addition, the development of the trailing vortices is not as great as the use of the flat and the concave shapes. However, the maximum value of the radial velocity component is found for the second configuration which represents the association of the flat blade at the top and the convex blade at the bottom ($u_{\max} = 0.31 U_{\text{tip}}$). The lowest value is found by using the concave blade instead of the convex blade, due to the high strain created by the interaction between the blade and the flow ($u_{\max} = 0.22 U_{\text{tip}}$). It can be seen that the blade at the bottom of the tank has the greatest effect on the flow. In fact, the maximum value of the radial velocity is almost similar while we use the flat blade turbine at the bottom ($u_{\max} = 0.27 U_{\text{tip}}$).

Figure 7 shows the distribution of the axial velocity component of the curved blade turbine. According to these results, the highest value region is localized at the bottom of the tank close to the blade that represents the suction of the flow of the blade. The second one is localized besides the wall of the tank above the blade at the same direction with the recirculation loops. Then, two lowest value regions were presented. In fact, the largest one is localized close to the free surface, while the

Figure 6.
Radial velocity for staged system.

Figure 7.
Axial velocity for one-staged system.

narrowed one is localized besides the wall of the tank at the bottom. The development of the maximum value of the axial velocity component is larger with the convex blade, which confirms that the recirculation loops associated to the convex blade are more important according to the other configurations. According to the velocity field, it can be seen that the recirculation loops associated to the convex blade are larger than the other configurations that are in conjunction with the amelioration of the axial velocity component with the convex blade. The highest maximum axial velocity component is generated by the flat turbine by $v_{\max} = 0.22 U_{\text{tip}}$ followed by the concave blade by $v_{\max} = 0.16 U_{\text{tip}}$ and by the convex blade by $v_{\max} = 0.15 U_{\text{tip}}$.

For the staged system (**Figure 8**), the maximum value of the axial velocity component is localized at the lowest blade that represents the suction of the flow. The highest maximum value is produced by the second configuration by $v_{\max} = 0.4 U_{\text{tip}}$. For the first configuration, the highest value is equal to $v_{\max} = 0.29 U_{\text{tip}}$. In addition, the maximum value of the axial velocity is almost similar while we use the flat blade turbine at the bottom ($v_{\max} = 0.33 U_{\text{tip}}$).

Figure 8.
Axial velocity for staged system.

4.3 Rms of velocity fields

Figure 9 shows the root mean square of the velocity field of the curved blade turbine. The root mean velocity presents the fluctuation of the periodic velocity. According to these results, the highest value region is localized at the same direction of the trailing vortices. In fact, for the convex configuration, the fluctuated velocity is localized close to the blade tip. For the flat blade turbine, the turbulent fluctuation is propagated to the vessel wall. For the concave blade, the fluctuation is localized at the upper and the downer edges of the blade. The maximum variability of the flow occurs due to the trailing vortices following the recirculation loops.

For the staged system (**Figure 10**), the greatest fluctuation is localized at the blade that is placed at the bottom of the tank. These fluctuations can be explained by the suction of the flow from the bottom of the tank. In addition, the maximum values of the turbulent fluctuations are created between the two blades due to the interaction between the blades. The fluctuations generated due to the association of the convex impeller with the flat impeller are narrowed when they are compared to the association of the concave and the flat impeller. This effect reveals that the

Figure 9.
Rms velocity for the one-staged system.

Figure 10.
Rms velocity for the staged system.

convex blade is not able to create a large fluctuation on the turbulent flow and local vortices are created.

4.4 Vorticity

Figure 11 shows the vorticity generated with the curved blade turbines. According to these results, the bulk region of the tank is presented with the medium vorticity value. For the flat and the concave configurations, the propagation of the vorticity is larger than the convex blade. The highest value area is localized in the upper region of the tank which follows the first circulation loop at the same direction of the von Kármán vortex street. The lowest value area is localized in the inferior region of the tank which follows the same direction of the second circulation loop. In fact, it has been noted that the highest recirculation loops are more energetic than the lowest ones. For the convex blade two maximum regions are created presenting the clockwise and the counterclockwise (CW-CCW) vortex pair at the blade tip.

For the staged system (**Figure 12**), the vortical structures are localized at the region between the two blades at the same direction of the discharge flow of each blade, which explains the domination of the trailing vortices at the turbulent flow.

Figure 11.
Vorticity for the one-staged system.

Figure 12.
Vorticity for the staged system.

Figure 13.
Dimensionless turbulent kinetic energy distribution of the one-staged system.

4.5 Turbulent kinetic energy

Figure 13 shows the distribution of the turbulent kinetic energy of the curved blade turbine. The turbulent kinetic energy is dimensionless by the square of the tip velocity. According to these results, the turbulent kinetic energy is maximum at the blade tip, and it decreases progressively moving away from the blade. As it was found in the previous sections, the convex blade dissipates the highest energy in the flow. The energy produced by the flat blade decreases by 37% and by 27% for the concave configuration. For the staged system (**Figure 14**), the maximum turbulent kinetic energy is localized between the two blades at the same direction as the trailing vortices. In addition, the turbulent kinetic energy is larger at the association of the concave blade than the convex blade.

4.6 POD analysis

In this section, we used the decomposition of the flow basing on the eigenvalues. This method allows to reveal the smallest vortical structure that cannot be seen by the usual mean flow according to its energetic amount by using the dimensionless eigenfunction. In fact, many vortices are presented with different sizes and shapes. For the one-staged system (**Figures 15–17**), the loop created at the blade tip is the most energetic. The largest one is obtained from the flat turbine. However, the

Figure 14.
Dimensionless turbulent kinetic energy distribution of the staged system.

Figure 15.
POD field for the flat blade.

Figure 16.
POD field for the concave blade.

narrowed loop is created by the concave shape, which can be explained by the axial velocity above the impeller. The clockwise and the counter clockwise (CW-CCW) vortex pair at the blade tip can be clearly seen at the highest modes that look similar

to the development of the trailing vortices of the Rushton turbine [18, 19]. The trailing vortices are more extended by using the concave blade. However, the flow reaches the bottom of the tank faster by using the convex form.

Figure 17.
POD field for the convex blade.

Figure 18.
POD field for the PDh, PI concave.

Figure 19.
POD field for the PDh, PI convex.

Figure 20.
POD field for the PDb, PI concave.

Figure 21.
POD field for the PDb, PI convex.

For the staged systems (**Figures 18–21**), it can be seen that many vortices are created at the region localized between the two blades. This region represents the interaction between the highest and the lowest impeller. Hence, the flow becomes more energetic that explain the cause of the development of these vortices. The flow reach the free surface as well as bottom of the tank faster by mounting the flat turbine at the top of the tank. The trailing vortices become more energetic by using the combination between the flat blade at the bottom and the convex blade at the top.

The development of the different modes shows that the shoes of the combina- tion is extremely important and can affect the mixing inside the vessel. Conse- quently, the combination between impellers can lead to affect the final product in terms of homogeneity and the cost in terms of the time mixing and power con- sumption. This found contradicts what has been observed in the study of the mean velocity field, as it gives almost the same results. In addition, it proves that the mean flow is not able to show the real behave of the flow.

5. Conclusion

The objective of this paper is to investigate experimentally the hydrodynamic structure of the curved blade turbine using the particle image velocimetry. Thereby,

several results were evaluated which contain velocity field, axial and radial velocity distribution, root mean square velocity, vorticity, and the turbulent kinetics energy. Two circulation loops were presented. The jet flow is more intensive for the convex blade turbine. However, the concave configuration produces a larger lowest loop than the other configurations. Hence, the downer region of the tank is more turbulent than the flat and the convex configurations. The maximum radial velocity generated by the flat blade turbine spreads to reach farther places. This can be explained by the ability of the blade shape to generate training vortices. The convex shape of the blade gives the turbine the ability to move easily within the water and transmit more velocity and energy while not giving it enough capacity to expand much. In fact, it has been noted that the convex blade is not able to create a large fluctuation on the turbulent flow and local vortices are created. In addition, it has been noted that the fluctuation of the flow is dominated by the trailing vortices more than by the recirculation loops. For the staged system, an oblique flow is created between the two impellers, and turbulent fluctuations are greater at this region due to the interaction between the blades.

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