CFD simulation of the laminar flow in stirred tanks generated by double helical ribbons and double helical screw ribbons impellers

Research article

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Abstract: In this paper, the mixing performance of double helical ribbons and double helical screw ribbons impellers mounted on stirred tanks is numerical investigated. The computer simulations are conducted within a specific computational fluid dynamic (CFD) code, based on resolution of the Naviers-Stokes equations in the laminar flow with a finite volume discretization. The field velocity and the viscous dissipation rate are presented in different vessel planes. The global characteristics and the power consumption of these impellers are also studied. The numerical results showed that the velocity field is more active with the double helical screw ribbons impeller. In this case, the effectiveness of the viscous dissipation and the pumping flow has been obviously noted. Also, the pumping and the energy efficiency reach the highest values at the same Reynolds number. The good agreement between the numerical results and the experimental data quietly confirmed the analysed method.

Keywords: CFD • Finite volume • Modelling • Laminar flow • Double helical ribbons • Screw • Stirred tank
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1. Introduction

Viscous dispersive mixing is an important unit operation in polymerization, food and other industrial processes. To respond to the needs of industrial processes, various impellers have been developed and a number of studied various impeller characteristics have been reported; useful reviews have been given in Nagata [1]. Among the different impellers available, the helical ribbon and the screw

impellers are considered to be more efficient for the agitation of highly viscous liquids. In the literature, many works have devoted to these impellers. For example, Kaneko et al. [2] analysed the three-dimensional motion of particles in a single helical ribbon agitator by the discrete element method. To validate the computed results, experiments were carried out with a cold scale model of 0.3 m inside diameter. Circulation time of particles in the agitator and the horizontal particle velocity distribution in the core region predicted by the simulation agreed well with those obtained by experiments. Niedzielska and Kuncewicz [3] discussed experimental investigations and model calculations of power consumption and heat transfer coefficients

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from a vessel wall to mixed liquid for ribbon impellers operating in a laminar regime. On the basis of the obtained results, the heat transfer efficiency of particular impellers was calculated and geometrical dimensions of the most efficient ribbon impeller were determined. Masiuk and Rakoczy [4] showed that the theory of information can be used to describe the random process of mixing of granular materials in a multi-ribbon blender. A new form of the entropy criterion estimating the current state of a mixture is proposed, a mathematical model to describe the variations of the informational entropy during the process is developed, and the experimental validation of the model is done for the blending process duration. Anne-Archard et al. [5] investigated the hydrodynamics and power consumption in laminar stirred vessel flow using numerical computation. The Metzner-Otto correlation was established for mixing in power-law fluids. Distributions of shear rates and their link to power consumption for helical and anchor agitators are discussed. Depending on the fluid model, the concept is valid or constitutes a satisfactory approximation for fully sheared flows. Iran-shahi et al. [6] compared the viscous mixing characteristics of the Ekato Paravisc with those of an anchor and a double helical ribbon. The methodology is based on 3D CFD finite element-based simulations. The predictions are validated firstly by comparing the Newtonian and non-Newtonian power consumptions and mixing times against literature experimental data. Then, the computed 3D laminar flow patterns and several mixing performance criteria of the impellers are investigated such as power consumption, pumping capability, intensity of segregation, mixing time, mixing efficiency and specific energy. Estelle and Lanos [7] studied rheological behaviour of fluids in mixing systems using a simplified approach in shear rate calculation. The efficiency of this approach is investigated from Newtonian and non-Newtonian fluids, exploring also the geometry of mixing systems. Results are favourably compared with those obtained from previous published works. With helical ribbon, they have considered the virtual radius of the probe obtained by Roos et al. [8]. The use of the real radius leads to an error in calculated shear stress. This means that the fluid is mainly sheared at the inner periphery of the helical ribbon as the virtual radius is closed to the inner radius of this geometry. Seyssiecq et al. [9] devoted to the triphasic in situ rheological characterisation of an activated sludge, with total suspended solid and operated in a bioreactor under different stirring and aeration rates. The bioreactor is a Plexiglas vessel of diameter 0.15 m. It is equipped with a double helical ribbon impeller rotating rather close to the wall. In this work, the flow properties are directly measured inside the bioreactor. Muzzio et al. [10] investigated the effects of processing and equipment parameters of a rib-

bon blender on magnesium stearate homogeneity. A core sampling technique is used to obtain at least one hundred samples per sampling event, which are extracted throughout the blender and yield a thorough characterization of the entire bed. Maingonnat et al. [11] studied the power consumption of a double ribbon impeller as a function of the operating conditions and the rheological behaviour of the experimental fluids. The isothermal experiments carried out with newtonian fluids gave the power curve for purely laminar and transitional flow regimes and the Kp factor was determined for the laminar flow regime. The isothermal experiments with two shear thinning fluids and the application of the Metzner and Otto concept led to the determination of the Ks factor [12]. The experimental values of the Kp and Ks factors are in a close agreement with previous works carried on this type of equipment. Using the multiple reference frames impeller method, Zhang et al. [13] simulated the three-dimensional non-Newtonian flow field generated by a double helical ribbon impeller. The velocity field calculated by the numerical simulation was similar to the previous studies and the power constant agreed well with the experimental data. A new method was devised in this paper to use the area-weighted average viscosity around the impeller as the effective viscosity for calculating Ks. This method showed both good accuracy and ease of use. Riveraa et al. [14] studied the macro-mixing mechanisms of the Super blend coaxial mixer consisting of a Maxblend impeller and a double helical ribbon agitator mounted on two independent coaxial shafts rotating at different speeds. To model the rotation of agitators a hybrid approach based on a novel finite element sliding mesh and fictitious domain method is used. The power consumption, the flow patterns, the shear rate distribution, the pumping capacity and the mixing time of the Super blend mixer are calculated from the simulated hydrodynamics. In this work, the Super blend coaxial mixer is found as a good alternative for tough mixing applications. Barbot et al. [15] studied the effect of NaN₃ addition on both volumetric oxygen mass transfer coefficient and rheology of activated sludge suspensions in a bioreactor equipped with a double helical ribbons impeller. Yu et al. [16] studied mixing performance in high solid anaerobic digester with A-310 impeller and helical ribbon. A mathematical model was constructed to assess flow fields. A systematic comparison for the interrelationship of power number, flow number and Reynolds number was simulated in a digester with less than 5% and 10% total solids. The simulation results suggested a great potential for using the helical ribbon mixer in the mixing of high solids digester.

The literature review proves that the double helical ribbons impeller is always used for the mixing operation and particularly for the highly viscous liquid. As a conse-

quence, it appears important to study this impeller and to make a comparison with the double helical screw ribbons impeller through a CFD simulation. The numerical results presented in this paper constitute for us an essential stage permitting to validate the numerical method developed in our CFD code. In fact, it is the first version of the used code date since twenty years [17, 18]. The update version is now able to simulate automatically different impellers. The numerical method called "Meshing adaptation to the solid shape" is recently developed and used to study different impellers. For further information about this method, the lectures can be referred to the new researches paper [19, 20]. In our simulation process, we have begun with the simple shape of the turbine like the flat-blade turbine [21, 22]. After that, the situations have been complicated and we have studied others impellers presenting a complex shape. Particularly, we are interested to the laminar or turbulent flow generated by the retreated-blade turbines [21], the pitched-blade turbines [19], the modified pitched-blade turbines [23], the Rushton turbines [24], the Helix turbines [25], the Ancre impellers [26, 27] and the screw impeller [20].

In this new work, we are interested to compare the laminar flow results of the double helical ribbons impeller with the double helical screw ribbons impeller in agitator vessel. The geometric arrangement considered is not numerically analysed elsewhere.

2. Geometric Arrangement

Figure 1 shows the details of the stirred tanks equipped by double helical ribbons and double helical screw ribbons. The first impeller (Figure 1a) has the same characteristics defined by the Nagata application [1]. The second is an association of a double helical ribbons and a screw impeller (Figure 1b). These impellers dimensions are defined by $b=0.1d$ and $h=d$. The tip to tip impeller diameter ratio d/D is a 0.95. The shaft is placed concentrically with a diameter ratio s/D of 0.04. In this arrangement, D is the diameter of the cylindrical stirred tank having a plated bottom with a height-to-diameter ratio H/D of 1.

3. Model formulation

3.1. Mathematical modeling

The simulation of the laminar flow field of the double helical ribbons and the double helical screw ribbons impellers in a stirred tank is governed by continuity and Navier-Stokes equations. The continuity equation for in-

Figure 1. Stirred tank equipped by proximity impellers.

compressible fluids is given in the following form:

$$\text{div } \vec{V} = 0 \quad (1)$$

Navier-Stokes equations are written in a rotating frame reference. Therefore, the centrifugal and the Coriolis accelerations terms are added [28]. These equations, written in cylindrical coordinates (r, θ, z) , are expressed in the general conservation form which can be written as follows:

$$\frac{\partial(\rho \vec{V})}{\partial t} + \text{div}(\rho \vec{V} \otimes \vec{V} + p \vec{I}) = \text{div}(\bar{\tau}) + \rho \left(\Omega^2 \vec{r} - 2\vec{\Omega} \wedge \vec{V} \right) \quad (2)$$

The equations system is solved using the three-dimensional CFD code developed in our Laboratory [19–28]. The dimensional analysis enables us to characterize power consumption in a stirred tank through the power number N_p [29] defined as follows:

$$N_p = \frac{P}{\rho N^3 d^5} \quad (3)$$

In case of the laminar flow, the total power consumption was calculated from the general relationship:

$$P = \int_{V_c} \mu \Phi_v dv \quad (4)$$

The viscous dissipation function Φ_v can be expressed in cylindrical coordinates in the following form:

$$\Phi_v = \left[2 \left[\left(\frac{\partial U}{\partial r} \right)^2 + \left(\frac{\partial V}{r \partial \theta} + \frac{U}{r} \right)^2 + \left(\frac{\partial W}{\partial z} \right)^2 \right] + \left[\frac{\partial V}{\partial r} - \frac{V}{r} + \frac{\partial U}{r \partial \theta} \right]^2 + \left[\frac{\partial W}{r \partial \theta} + \frac{\partial V}{\partial z} \right]^2 + \left[\frac{\partial U}{\partial z} + \frac{\partial W}{\partial r} \right]^2 \right] \quad (5)$$

The pumping flow number was calculated using the following equation:

$$NQ_p = \frac{Q_p}{N d^3} \quad (6)$$

The numerical data of mean axial velocity below the impeller, in the radial direction were used for the calculation of pumping flow Q_p :

$$Q_p = \int_{z_B}^{z_B+h} \pi d (U)_{r=d/2} dz + \int_0^{d/2} 2\pi r (W)_{z=z_B} dr \quad (7)$$

The pumping efficiency [23] can be defined by dividing the pumping flow number by the power number:

$$E_p = \frac{NQ_p}{N_p} \quad (8)$$

However, the energy efficiency can be obtained as follows:

$$E_e = \frac{NQ_p^3}{N_p} \quad (9)$$

3.2. Numerical method

Our computational fluid dynamics (CFD) code is based on solving the continuity and the Navier-Stokes equations using a finite volume method. The transport equations are integrated over its own control volume using the hybrid scheme discretization method. The discretized equations were solved iteratively using the SIMPLE algorithm for pressure-velocity coupling [30]. The algebraic equation solutions are obtained in reference to the fundamental paper published by Douglas and Gunn [31]. The discretization method and numerical solution procedure used have been described in detail elsewhere [19–28].

For the meshing, we used the design software SolidWorks to construct the impeller shape. Then, a list of nodes belonging the interfacing separating the solid domain of the flow domain was defined. Using this list, the meshes in the flow domain are automatically generated for the three-dimensional simulations. Therefore, the region to be modeled is subdivided into a number of control volumes defined on a cylindrical coordinates system (r, θ, z). A staggered mesh is used in such a way that four different control volumes are defined for a given node point, one for each of the three vector components and one for the pressure. The flow field was computed using a grid size of $NR=30$, $N\theta=60$ and $NZ=60$. The grid elements can affect the quality of the solution. In fact, a preliminary grid convergence study was carried out in order to verify that the solution is a grid independent. For different

types of meshes, we are interested to compare the numerical results with experimental data. The number of grid nodes in both the inner and outer mesh zones was systematically increased throughout the vessel. Generally, the numerical simulation with a coarse grid has a large deviation from the reference results. Gradually refining the grid, we note that the appearance of results stabilizes and approaches to the reference values. Also, it's noted that it is unnecessary to further refine the mesh, because the calculation time becomes very large and the results found are almost the same. Thus, we show that there is a compromise between the mesh refinement and choice of computing time.

To simplify calculation, a steady flow field on the rotating frame fixed on the impellers is adopted. In these conditions, a no-slip condition on the non-moving impeller and a rotational speed on the tank walls are considered. In order to take into account the presence of the impeller, all radial as well as tangential and axial velocity mesh nodes, which intersect with impeller, were taken equal to zero. However, at the internal wall we have to set the angular velocity component equal to the rotating speed because of the rotating frame. In the part of the boundary where the fluid leaves the computation domain, zero velocity gradients are assumed. The solution was obtained when the total residuals for the equations dropped to below 10^{-6} .

4. Numerical results

Our computer simulations results, such as the flow patterns, the viscous dissipation rate and the evolution of the pumping flow number NQ_p , the pumping efficiency number E_p , the energy efficiency E_e the power number N_p , offer local and global information about the laminar mixing within double helical ribbons and double helical screw ribbons impellers mounted on stirred tanks. They give a more precise understanding of the hydrodynamic mechanism than those obtained by experimental studies. The flow conditions are represented by the Reynolds number equal to $Re=4$ and the Froude number equal to $Fr=0.087$. These results obtained by our code are compared with the ones found by Nagata application [1].

4.1. Flow patterns

4.1.1. Flow patterns in $r-\theta$ plane

Figure 2 shows a velocity vector plot of the primary flow (U, V) presented in $r-\theta$ plane defined by the axial coordinate equal to $z=1$. It appeared that the flow was strongly dominated by the tangential component. Far from the re-

Figure 2. Flows patterns induced in r - θ plane defined by $z=1$.

gion swept by the double helical ribbons impeller, it has been noted a progressive slowing of the flow. Also, it has been noted that the velocity vector was directly affected by the proximity impellers type. Indeed, within the double helical screw ribbons impeller, it has been observed that the velocity field is very active in the two opposite sides that are confounded with the impeller tip.

4.1.2. Flow patterns in r - z planes

Figures 3, 4 and 5 show a velocity vector plot of the secondary flow (U, W) presented in different r - z planes. These presentation planes have been defined respectively by the angular coordinate equal to $\theta=122^\circ$, $\theta=136^\circ$ and $\theta=316^\circ$. These positions have been chosen in order to show the velocity field evolution. With a double helical ribbons impeller, it's noted that the flow have a centrifugal radial movement in the middle of the tank in the first plane (Figure 3a). This movement decrease in the second plane (Figure 4a). However, it becomes a centripetal type in the third plane (Figure 5a). In the swept domain localised in the top of the tank, the flow have a descending axial movement. However in the bottom, this movement reverses and becomes an ascending and oblique character. While approaching to the lateral surface of the tank, the flow becomes weak. Within a double helical screw ribbons impeller, the same observations are noted. But, it's clear that the velocity field is more active than the double helical ribbons impeller.

4.1.3. Radial profiles of the dimensionless velocity components

Figures 6, 7 and 8 illustrate the predicted radial profiles of the dimensionless radial $U(r)$, tangential $V(r)$ and axial $W(r)$ velocity components of the double helical ribbons and the double helical screw ribbons impellers. These profiles are presented in two different r - θ planes defined by the dimensionless axial coordinates equal to $z=0.45$

Figure 3. Flows patterns induced in r - z plane defined by $\theta=122^\circ$.

Figure 4. Flows patterns induced in r - z plane defined by $\theta=136^\circ$.

Figure 5. Flows patterns induced in r - z plane defined by $\theta=316^\circ$.

Figure 6. Radial profiles of the radial velocity $U(r)$.

and $z=0.8$. These figures adequately portray the swirling radial, tangential and axial jets character. In these figures, the numerical results of the two impellers were superposed to compare the local characteristics. Figure 6a shows that the radial velocity component $U(r)$ reaches its maximal value equal to $U=0.036$ in the horizontal plane situated in the tank bottom. The radial position corresponding to this maximal value is defined by $r=0.33$. This result is also observed for tangential $V(r)$ and axial $W(r)$ velocity components. In fact, the tangential velocity component $V(r)$ reaches its maximal value $V=0.3$ in the radial position equal to $r=0.5$ (Figure 7a). Whereas, the axial velocity component $W(r)$ reaches its maximal value $W=0.03$ in the radial position equal to $r=0.58$ (Figure 8a). The minimal value $W=-0.012$ is reached in the radial position equal to $r=0.16$ (Figure 8b). The negative value of the axial component corresponds to a downward movement toward the tank bottom. In the tank top, it's noted a progressive reduction of the flow movements. In fact, the radial movement decreases and the radial velocity component $U(r)$ reaches very weak values. The tangential velocity component $V(r)$ shows a parabolic pace. It reaches very low values in the neighborhood of the axis and the tank walls. For the axial velocity component $W(r)$, it's noted a more intense downward movement.

Figure 7. Radial profiles of the tangential velocity $V(r)$.

Figure 8. Radial profiles of the axial velocity $W(r)$.

Figure 9. Dissipation rate induced in r - θ plane defined by $z=0.1$.

Figure 11. Dissipation rate induced in r - θ plane defined by $z=1.15$.

Figure 10. Dissipation rate induced in r - θ plane defined by $z=0.95$.

Figure 12. Dissipation rate induced in r - θ plane defined by $\theta=108^\circ$.

4.1.4. Viscous dissipation rate

Figures 9, 10 and 11 show respectively the viscous dissipation rate in different r - θ planes defined respectively by the axial positions equal to $z=0.1$, $z=0.95$ and $z=1.15$. Figure 12 shows the viscous dissipation rate in r - z plane defined by the angular position equal to $\theta=108^\circ$. Seen the symmetry of the problem, two symmetry wake shape are observed in these r - θ planes. Globally, the maximal value of the viscous dissipation rate is reached in the meeting of the double helical ribbons with these presentation planes. Out of this domain, the viscous dissipation rate becomes rapidly very weak. All of these observations are available for the two impellers. But, it's noted that the viscous dissipation rate for the double helical screw ribbons impeller is more important than the double helical ribbons impeller.

4.2. Global characteristics

To compare the global characteristics of the double helical ribbons in closed stirred tanks to the double helical screw ribbons, the pumping flow number NQ_p , the pumping efficiency number E_p and the energy efficiency E_e are calculated from the CFD code. The dependence of these

characteristics on Reynolds number Re was presented respectively in Figures 13, 14 and 15. Globally, these parameters were proportional to the Reynolds number Re in the laminar flow regime. At the same Reynolds number, it's clear that the double helical screw ribbons impeller characteristics values are superior to the double helical ribbons impeller. According to these results, we can conclude that the association of the double helical ribbons impeller with the screw impeller can improve the mixing operation. This fact can be used for the impeller design.

5. Comparison with experimental results

Figure 16 shows the power number N_p variation on Reynolds number Re in the laminar flow range of the double helical ribbons and the double helical screw ribbons impellers. In these conditions, the power number was inversely proportional to the Reynolds number and the value

Figure 13. Pumping flow number variation $NQ_p = f(Re)$.

Figure 15. Energy efficiency variation $E_e = f(Re)$.

Figure 14. Pumping efficiency number variation $E_p = f(Re)$.

Figure 16. Power number variation $N_p = f(Re)$.

of the product $K_p = N_p Re$ remained constant. According to these results, it's noted that these curves present the same linear variation. Moreover, it's noted that the power number of the double helical screw ribbons is superior to the double helical ribbons at the same Reynolds. To verify our computer results, the power number calculated from the CFD code were compared with the experimental results found in the literature in the case of the double helical ribbons. In these conditions, the numerical value of the power number N_p is slightly inferior to the experimental value when compared at the same Reynolds number Re . These experimental results founded by Nagata [1] were superposed over an average error of 10%. The good agreement between the experimental results and the numerical results quietly confirmed the analysis method.

6. Conclusion

Using our specific Computational Fluid Dynamics (CFD) code, three-dimensional simulation of the laminar flow generated by the double helical ribbons and the double

helical screw ribbons impellers was investigated. Numerical results concerning velocity fields, viscous dissipation rate and global characteristics are presented in this paper. These results showed that the velocity field is more active with the double helical screw ribbons impeller. In this case, the effectiveness of the viscous dissipation and the pumping flow has been obviously noted. The pumping and the energy efficiency have reached the highest values at the same Reynolds number. The comparison of the power number has been presented to be compared with ones found by other researchers. The good agreement between the numerical results and the experimental data validate the numerical method. In the future, we intend to characterise the hydrodynamic structure of this impeller by the particle image velocimetry (PIV) technique.

Nomenclature

d impeller diameter, m
 D internal diameter of the vessel tank, m
 E_e energy efficiency, dimensionless

E_p	pumping efficiency number, dimensionless
Fr	Froude number, dimensionless, $Fr = \frac{(2\pi N)^2 d}{g}$
g	gravity acceleration, $m^2 s^{-1}$
h	impeller height, m
H	vessel tank height, m
N	rotation speed of the impeller, s^{-1}
N_p	power number, dimensionless
NQ_p	pumping flow number, dimensionless
P	power, W
p	pressure, Pa
Q_p	pumping flow, $m^3 s^{-1}$
Re	Reynolds number, dimensionless, $Re = \frac{\rho N d^2}{\mu}$
r	radial coordinate, dimensionless, $r = \frac{r^*}{d}$
r^*	dimensional radial coordinate, m
s	shaft diameter, m
t	time, s
U	radial velocity components, dimensionless, $U = \frac{U^*}{2\pi N d}$
U^*	dimensional radial velocity components, ms^{-1}
V	angular velocity components, dimensionless, $V = \frac{V^*}{2\pi N d}$
V^*	dimensional angular velocity components, ms^{-1}
W	axial velocity components, dimensionless, $W = \frac{W^*}{2\pi N d}$
W^*	dimensional axial velocity components, ms^{-1}
z	axial coordinate, dimensionless, $z = \frac{z^*}{d}$
z^*	dimensional axial coordinate, m

Greek symbols

θ	angular coordinate, rad
μ	fluid viscosity, Pas
η	fluid viscosity, dimensionless
ρ	fluid density, kgm^{-3}
Φ_v	viscous dissipation rate, dimensionless
Ω	angular velocity, $rad s^{-1}$

Vector and tensors

$\vec{V}$	velocity vector
$\vec{\Omega}$	angular velocity vector
$\vec{I}$	identity tensor
$\vec{\tau}$	stress tensor
$\vec{\tau}_R$	Reynolds tensor

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