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ORIGINAL ARTICLE

Effects of turbulization on the disc pump performance

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Abstract Disc pumps are used for difficult pumping applications, such as, pumping of high suspension solids and abrasives, viscous fluids, air entrained and shear sensitive fluids. The pumping mechanism, based on the boundary layer effect and the viscous drag minimizes the contact between the pump and the fluid reducing the wear level; but the pumping mechanism itself makes its efficiency low in comparison with other pumps for similar applications. This research aims to increase the performance of this pump developing a new experimental study based on the turbulization of flow by the placement of turbulizers in the interdisc channel output. The variables involved are the angular velocity (ω) and the cross section shape of the turbulizers. Eight impellers were constructed and evaluated using as cross section shape of turbulizers: the triangular, circular, and square. The experimental results show that the creation of circulatory currents, according to the Kutta-Johkovsky theorem, contributes to the increase the efficiency and the head of the disc pump and the square cross section shape of the turbulizers offers the best results.

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1. Introduction

The Common factors that have been prevailed in the evolution of pumping viscous fluid technologies are the high technical complexity in its construction and the high degree of wear suffered by de aggressiveness of the fluid; increasing the investment, maintenance and repair costs. One of the solutions for this problem is the use of Disflo pump or Disc pump as a result of its mechanism, based on the boundary layer effect and the viscous drag, minimizes the contact between the pump and the fluid reducing the level of wear. In Fig. 1 is shown that disc pumps allow to reduce initial capital cost, maintenance and downtime compared to other pumps for similar applications and services [1]. Pacello and Hanas [2] express that disc pumps with a different mechanism of pumping based on the effect of boundary layer and viscous drag is easy to build with a negligible level of wear.

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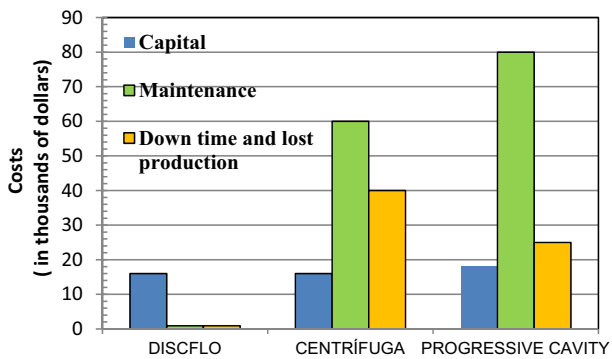

Fig. 1 Costs for different types of pumps in thousands of dollars. Take of [1].

Although the disc pump has a low level of vibration, with a consequent reduction in energy consumption and increasing the efficiency [3], this type of pump is less efficient than a similar centrifugal pump in non-viscous applications. The best efficiency point can often fall below 30 percent; therefore, the disc pump is not used for these applications, unless that the fluid has sensitivity to shear, high level of entrained air, severely abrasive content or frequent clogging by solids. The disc pump low efficiency as a result of the inefficiency in energy transfer for the viscous drag process and the pump clearances [4].

One factor that hinders the implementation of this technology in the viscous fluids pumping, is constraint in terms of head and efficiency to meet the requirements of the systems [5]. Pascoa and Oliveira [6] affirm report, that a micro scale disc pumps cannot compete in efficiency with impellers and therefore relegated to special applications. In the disc pump the diffuser volute must handle the flow with a very small entering angle, giving as result an inefficient volute. This is one of the reasons why viscous disc pumps have much lower efficiencies that might be expected from considerations of the flow in the rotor [7].

There are some methods for increasing the head and the efficiency of disc pump [8,9]. This paper shows the results of a theoretical and experimental study patented [10] which consist in placing eight turbulizers of different cross section to assemble the discs. These elements create circular currents, that increase energy transferred to the fluid from impeller; in this research different forms of the cross section will be used to see their effect on the performance of the pump. Due to the effect of viscosity, the boundary layer is separated producing high flow rates in the faces tangent to the radial direction and low flow rates in the faces perpendicular to the radial direction, generating pressure gradients in order to balance the energy [11].

The results show that the proven method is a way to increase the efficiency of this type of pump.

The shape of the cross section of the least hydrodynamic turbulence is the one that allows the best effect in the creation of circulatory currents.

2. Disc pumps characteristics

Disc pumps operation is based on the principles of boundary layer and viscosity drag. The main difference between the disc assembly and a standard centrifugal pump is that with the disc assembly, the pumpage does not impinge on the pump rotating mechanism and a pulsation-free laminar flow pattern through the pump is generated [1]. This feature works properly in disc pumps used in hard-to-pump applications such as pumping highly abrasive slurries, slurries with a high solid content, fluids with entrained air, and fluid containing delicate or shear sensitive product. The disc pump has a lower pumping efficiency than a similar centrifugal pump and therefore is not marketed for non-viscous applications. The disc pump also uses the same type of packing and seal arrangements as centrifugal pumps. A typical disc pump assembly is presented in Fig. 2 [2].

3. Experimental design

3.1. Design of the experiment. Descriptions and fundamentals

The design factors involved in the experiment are: X_1: angular velocity and X_2: shape of turbulizers. It is used to complete factorial design 2×4. The two levels of X_1 are related with the operation conditions of the case study $X_1\text{min} = 183\ S^{-1}$ and $X_1\text{max} = 293\ S^{-1}$. These angular speed values correspond with the characteristic values of rotation speeds of the disc pumps [1].

The four shapes of turbulizers are: circular (●), triangular with one vertex oriented toward the center of the disc (▼), triangular with one vertex oriented toward outside of the disc (▲) and square shape (■).

The selection of this shape is based on the flow over immersed bodies theory. All trials were done using eight turbulizers, the fluid used was honey of sugar mill process, keeping the cinematic viscosity in $v = 3.7$ St by temperature control in 52 °C according to the operational regimen. For a better understanding, a cross section view of the turbulizers placed on the periphery of discs is schematized in Fig. 3.

An impeller (Dispac) with square shape cross-section turbulizers and two intermediate discs is shown in Fig. 4. Turbulizers allow to separate the intermediate disks.

The two levels of angular velocity were inspected for every type and consequently, eight different results were obtained. The conformed experimental matrix is shown in Table 1.

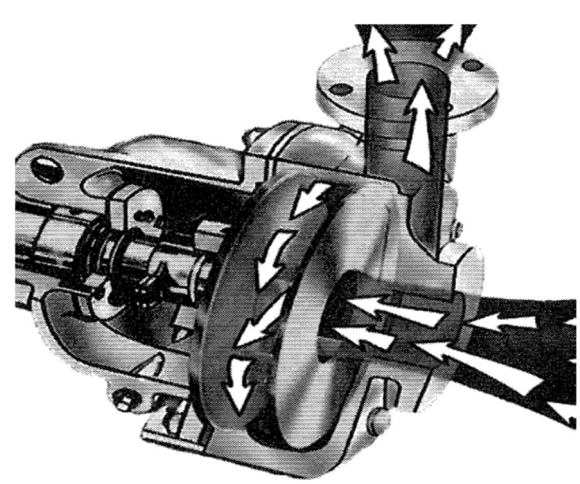

Fig. 2 Disc pump assembly.

4. Experimental setup

The experiments were performed in an experimental setup [12]. The variables measured and calculated during the experimentation are presented in Table 2.

The discharge and suction pressure was measured with a manometer and a vacuometer respectively; both of class 0.6. For flow measurement volumetric method was used. Baffle plates were used for damping the disturbance of discharged fluid. The electrical and power quality parameters were measured with an energy analyzers Fluke 435 II [13]. The output power was obtained with the plate and catalog data. The efficiency curve was obtained as a function of the load factor. With this curve we obtain a model of the electrical power as a function of the output power [14,15].

The uncertainty of the flow, head and power measurements were estimated to be 0.65%, 0.53% and 0.44% respectively. The uncertainty for efficiency was calculated for the Eq. (1) [16,17]

$$f_\eta = \sqrt{f_Q^2 + f_H^2 + f_P^2}; \% \quad (1)$$

where

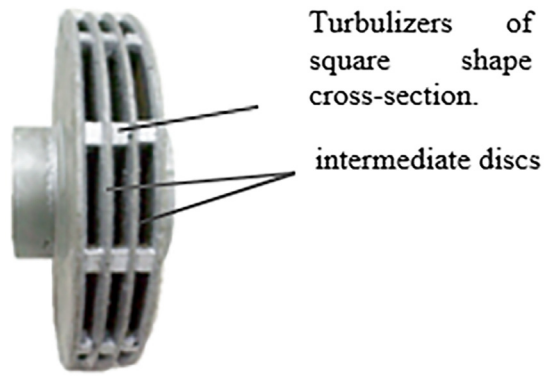

Fig. 4 Dispac with square shape cross section turbulizator and two intermediate discs.

f_η – Measurement uncertainty for efficiency; %
f_Q – Measurement uncertainty for flow; %
f_H – Measurement uncertainty for head; %
f_P – Measurement uncertainty for power; %

The value obtained is 0.94%.

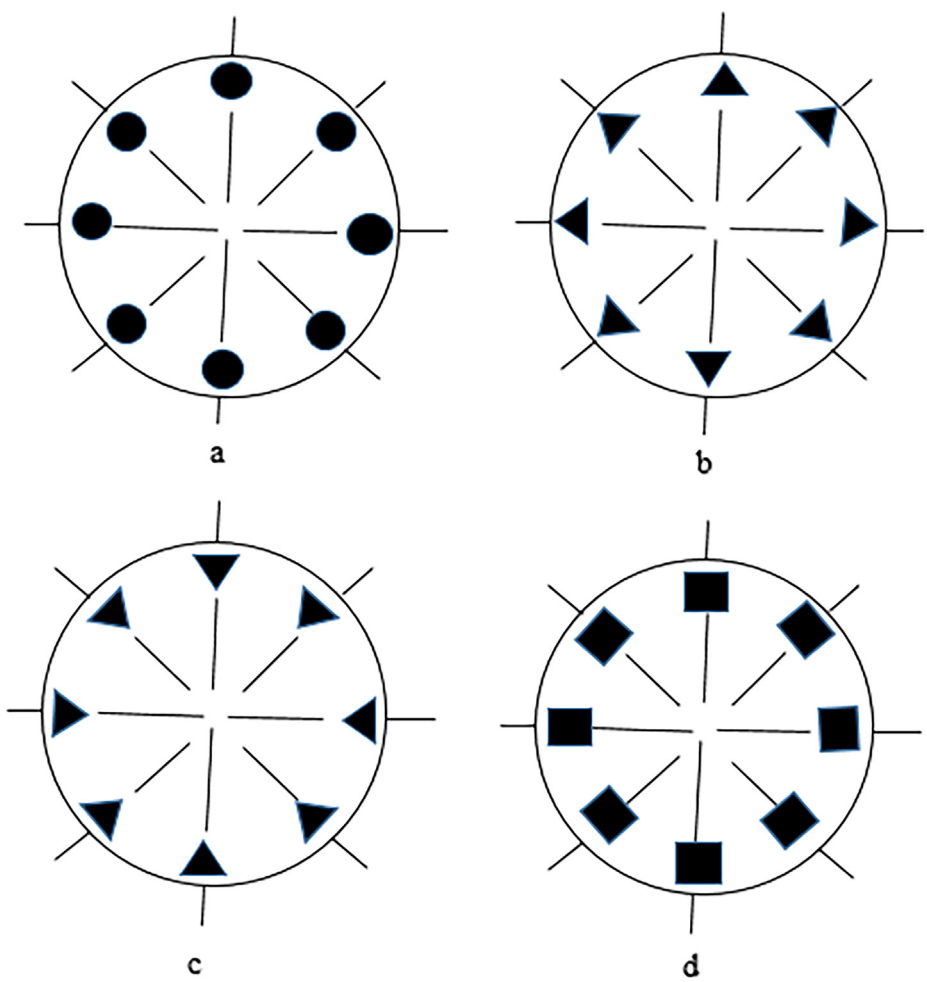

Fig. 3 Turbulizators shape: (a) circular, (b) triangular with one vertex oriented toward outside, (c) triangular with one vertex oriented toward the center, d) square.

Table 1 Experimental Matrix.

X_1 (s^{-1})	183	183	183	183	293	293	293	293
X_2	●	▼	▲	■	●	▼	▲	■

Table 2 Measured variables during experimentation.

Variable	Description
p_d	Discharge pressure [MPa]
p_s	Suction pressure [MPa]
Q	Pump flux[l/s]
P_m	Out Power [kW]
P_e	Electrical Power
I1 ,I2,I3	Electric current intensity for each phase

5. Results and discussion

The experimental information using a conventional algorithm according to Hydraulic performance acceptance tests, grades 1, 2 and 3 for pumps assessment, was processed according to ISO [18].

The pump head (H), flow (Q) and efficiency (Ef) values for the different cross-sectional shapes at two rotational speeds tested were obtained.

The results corresponding to the experiments performed are shown in Figs. 6–9. In Table 3 the regression models and degree of adjustment for head and efficiency for 183 s^{-1} rotational speed are shown.

It is important to highlight the level of correlation (R^2) between the experimental data and the given by the different models. The statistical analysis performed for a 95% confidence interval, at the angular velocity of 183 s^{-1} determine the difference among the mean values of the coefficients of regression models (Ef-Q) for different cross section shape of the of the turbulizers are shown in Table 4.

The results show significant differences between all the equations of adjustment Ef = f(Q) for the rotation speed ω_1. The difference is smaller comparing the shape of the triangular cross section with the vertex towards the center with the shape of the cross section with the vertex towards the periphery.

The influence of X_2 on H for X_1min and X_1max is represented in Figs. 5 and 6. In these figures it can be seen that the pump head and the flow cross increase when using square turbulizers.

When the angular velocity of the disc is increased, the influence of the cross section shape of the turbulizers on the head and the performance of the pump increases as well, while the field of flow velocities increases, it is obtained a greater positive effect of the circulation in the output impeller on the disc pump head.

When the disc pump operates with an angular velocity of 183 s^{-1} in the range of flows (Q) comprised between Q = 0 and Q = 0.78 l/s, the use of square turbulizers achieves a load increase in the range from 1.72 to 2.30 m in relation to the use of circular turbulizers.

Table 4 Statistical analysis for Efficiency-Flow regression models (Ef-Q). Angular speed ω_1 = 183 s^{-1}.

Comparison of regression models for different shape of the cross section	Difference between the mean values of the coefficients **Q**	Difference between the mean values of the coefficients $\mathbf{Q^2}$
● - ▲	−1.71 ± 0.12	0.51 ± 0.14
● - ▼	−1.58 ± 0.09	0.60 ± 0.12
● - ■	−2.62 ± 0.15	1.01 ± 0.17
▲ - ▼	0.13 ± 0.10	0.09 ± 0.11
▲ - ■	0.91 ± 0.16	0.50 ± 0.17
▼ - ■	1.04 ± 0.14	0.41 ± 0.15

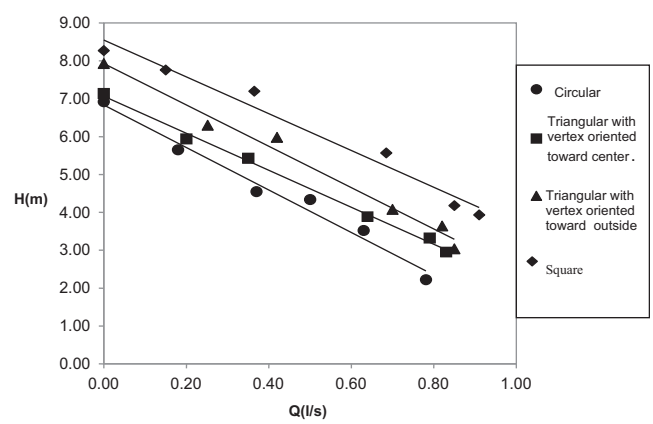

Fig. 5 Pump head curves for different turbulizers: ω_1 = 183S^{-1}.

Table 3 Regression Model for head and efficiency for ω_1 = 183S^{-1}.

Cross section shape	Regression models			
	Head (H),m; Flow (Q),l/s	R^2	Efficiency(Ef),%	R^2
Square (■)	H = −4.8543Q + 8.5464	0.9723	Ef = −5.2551Q2 + 7.02257Q	0.9885
Triangular with vertex oriented toward outside (▲)	H = −5.4553Q + 7.9275	0.9841	Ef = −4.75Q2 + 6.1067Q	0.9922
Triangular with vertex oriented toward center (▼)	H = −4.8821Q + 7.0642	0.9961	Ef = −4.8418Q2 + 5.9762Q	0.997
Circular (●)	H = −5.6021Q + 6.8311	0.979	Ef = −4.2438Q2 + 4.401Q	0.9801

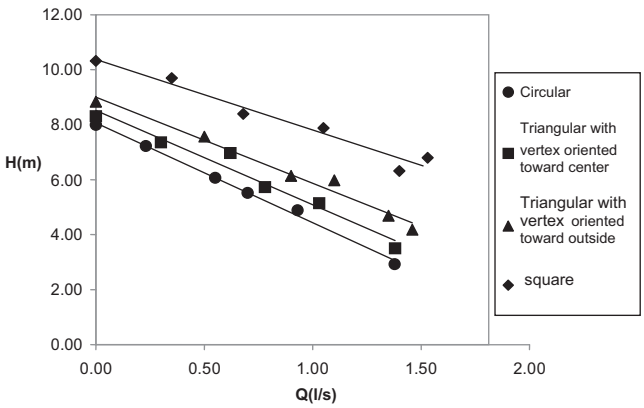

Fig. 6 Pump head curves for different turbulizers: $\varphi_2 = 293\ S^{-1}$.

In Figs. 7 and 8, it is observed that the resulting turbulization leads to an increased efficiency; it is more representative for upper values of angular velocity. This fact is explained with theory of flow over immersed bodies [19]. The shape of the cross section less hydrodynamic is that allows the best effect in the creation of circulatory currents.

Experiences made by the authors of this work to display the vortices created by the effect of placing turbulizers in the

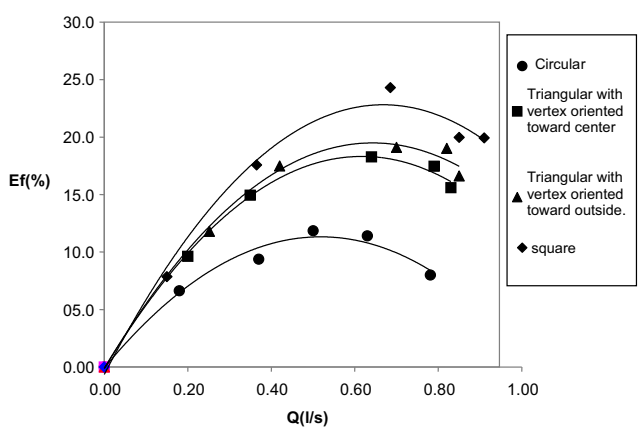

Fig. 7 Efficiency curves for different turbulizers: $\varphi_1 = 183S^{-1}$.

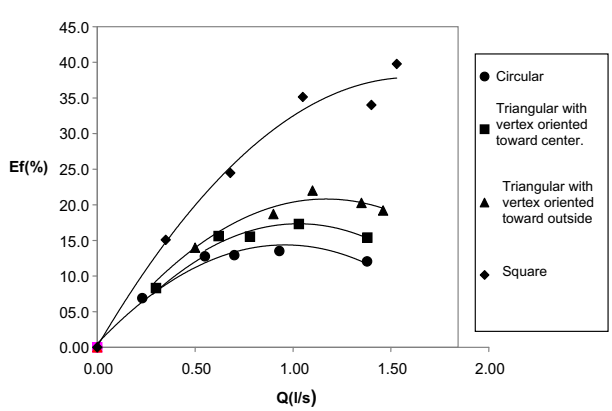

Fig. 8 Efficiency curves for different turbulizers shape: $\varphi_2 = 293\ S^{-1}$.

periphery of the dispac which rotates within a fluid can be seen in Fig. 9. These images are obtained through a stroboscopic lamp rotating at the same speed of the dispac to ($183\ s^{-1}$).

In Fig. 10 is observed a difference between the flow pattern that is originated in each case. The variation of turbulizers shape from circular shape to square shape increases eddies areas that contribute the creation of the flow circulation (Γ), increasing the head and the efficiency.

The turbulizers are centers for the creation of a model with a speed pattern characterized by vortex. For this model, Pennings, P., Westerweel, J. and Van Terwisga, T. [20] express that two parameters are sufficient to describe the velocity field. These are usually the viscous core radius r_v and the vortex circulation Γ. In the model mentioned above, the vorticity was strongly concentrated close to the vortex center. If the vortex circulation Γ in the outlet of the impeller increases, then the pump head increases.

6. Mathematical Argumentation

The mathematical argumentation of turbulization phenomenon to increase the disc pump head is treated for first time in this work. Apparently by means of the Physics – Mathematics laws it could be treated, but it is not possible for the following causes:

- Rupture of symmetry causes drawbacks that increase the degree of complexity of the solution.
- To represent physical reality, it is not necessary to make such a cumbersome analysis of the physical problem.
- The literature does not report an effective method to characterize this problem.

To demonstrate that the head of the disc pump increases with the inclusion of the turbulizer it is taken the expression given in [21].

$$H = \frac{W}{2\pi g}(\Gamma_2 - \Gamma_1) \tag{2}$$

where H – Head of the pump, (m); W – Angular velocity, (S^{-1}); $\Gamma 1$ and $\Gamma 2$ – vortex circulation at the inlet and the outlet impeller respectively, (m^2/s).

In Fig. 10 a schematic representation of the vortex circulation is shown.

Taking as reference the equation:

$$H = H_o + \Delta H \tag{3}$$

where

H – pump head with turbulizers.
H_o – pump head without turbulizers.
ΔH – head variation due to the inclusion of turbulizers.

If $\Delta H < 0$ the pump head decreases with the inclusion of turbulizers.

If $\Delta H > 0$ the pump head increases with the inclusion of turbulizers.

Substituting Eq. (1) in (2):

$$\Delta H = \frac{W}{2\pi g}[(\Gamma_2 - \Gamma_1) - (\Gamma_{20} - \Gamma_{10})] \tag{4}$$

Fig. 9 Circulation around turbulizers (a) circular (b) square.

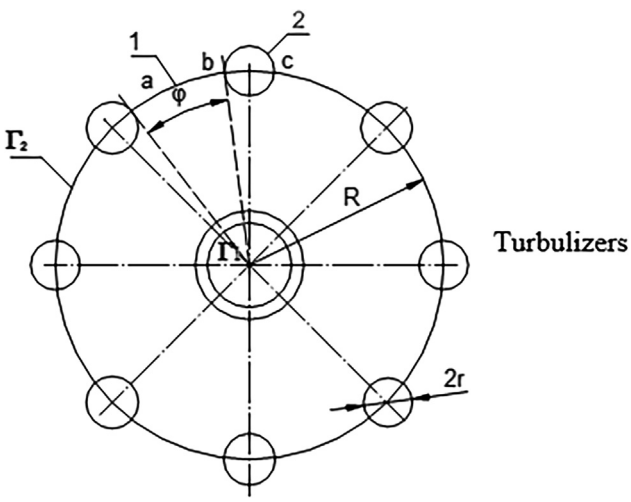

Fig. 10 Vortex circulation Γ_2, Γ_1 with turbulizators.

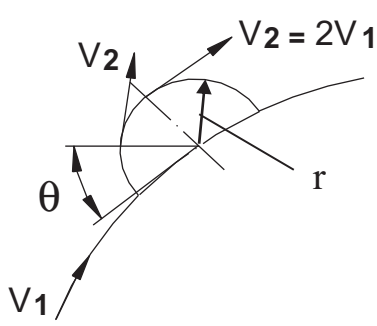

Fig. 11 Velocity around the turbulizers.

where

Γ_2, Γ_{20} –vortex circulation with and without turbulizers at the impeller outlet respectively.
Γ_1, Γ_{10} –vortex circulation with and without turbulizers at the impeller inlet respectively.

$\Gamma_1 = \Gamma_{10}$ since at the impeller inlet no turbulizers.

This leads to:

$$\Delta H = \frac{W}{2\pi g}(\Gamma_2 - \Gamma_{20}) \tag{5}$$

7. Calculation of circulation with turbulizers (Γ_2)

Circulation with turbulizers **(Γ_2)** is calculated by taking the schematic representation in Fig. 11.

Designating:

a-b as integration path 1.
b-c as integration path 2.

$$\Gamma_2 = \oint \vec{v} \cdot \vec{dl} = n\left[\int_1 \overrightarrow{V_1} \cdot \vec{dl} + \int_2 \overrightarrow{V_2} \cdot \vec{dl}\right] \tag{6}$$

$$\int_1 \overrightarrow{V_1} \cdot \vec{dl} = V_1 \cdot l_1 \tag{7}$$

where

l_1 – length of the section a – b or step between the turbulizers.
V_1– relative velocity at the impeller outlet.

The velocity V_2 is shown in Fig. 11. This corresponds to the velocity field around the turbulizers. It is assumed that turbulizers are cylindrical.

$$V_2 = 2 \cdot V_1 |\mathrm{sen}\,\theta|; \qquad 0 \leqslant \theta \leqslant \pi$$

Also

$$V_2 = 2 \cdot V_1 \cdot \cos\theta \qquad -\frac{\pi}{2} \leqslant \theta \leqslant \frac{\pi}{2}$$

$$\int_2 \overrightarrow{V_2} \cdot \vec{dl} = \int_2 2 \cdot \overrightarrow{V_1} \cdot \vec{dl} \cdot \cos\theta$$

$$\int_2 \overrightarrow{V_2} \cdot \vec{dl} = \int_{-\frac{\pi}{2}}^{\frac{\pi}{2}} 2 \cdot V_1 \cdot \cos\theta \cdot dl$$

$$dl = r \cdot d\theta$$

r – radio of the turbulizers

$$\int_2 V_2 \cdot \overrightarrow{dl} = r \int_{-\frac{\pi}{2}}^{\frac{\pi}{2}} 2 \cdot V_1 \cdot \cos\theta \cdot d\theta = 2 \cdot V_1 \cdot r \cdot sen\theta\Big|_{-\frac{\pi}{2}}^{\frac{\pi}{2}} = 4 \cdot V_1 \cdot r$$

$$\int_2 \overrightarrow{V_2} \cdot \overrightarrow{dl} = 4 \cdot V_1 \cdot r \tag{8}$$

Substituting Eqs. (6) and (7) in Eq. (5):

$$\Gamma_2 = n \cdot V_1(l_1 + 4 \cdot r) \tag{9}$$

The velocity without turbulizers is shown in the Fig. 12. The vortex circulation (Γ_{20}) in this case is given by:

$$\Gamma_{20} = \oint \overrightarrow{V_o} \cdot \overrightarrow{dl} = n\left(\int_1 \overrightarrow{V_{01}} \cdot \overrightarrow{dl} + \int_2 \overrightarrow{V_{02}} \cdot \overrightarrow{dl}\right) \tag{10}$$

where n-number of turbulizers

The velocity $\overrightarrow{V_{01}} = \overrightarrow{V_1}$

$$\int \overrightarrow{V_{01}} \cdot \overrightarrow{dl} = V_1 \cdot l_1 \tag{11}$$

$$\int_2 \overrightarrow{V_{02}} \cdot \overrightarrow{dl} = \int_2 V_1 \cdot dl \cdot \cos\alpha = \int V_1 \cdot r \cdot d\theta \cdot \cos\alpha;$$

$$0 \leqslant d\theta \leqslant \pi; \quad -\frac{\pi}{2} \leqslant \alpha \leqslant \frac{\pi}{2}$$

$$\int V_1 \cdot r \cdot d\theta \cdot \cos\alpha = \int V_1 \cdot r \cdot \cos\alpha \cdot d\alpha = V_1 \cdot r \cdot \mathrm{sen}\alpha\Big|_{-\frac{\pi}{2}}^{\frac{\pi}{2}}$$

$$\int \overrightarrow{V_{02}} \cdot \overrightarrow{dl} = 2V_1.r \tag{12}$$

Substituting Eqs. (10) and (11) in Eq. (9):

$$\Gamma_{20} = n \cdot V_1(l_1 + 2r) \tag{13}$$

Finally

$$\Gamma_2 - \Gamma_{20} = n \cdot V_1(l_1 + 4r) - n \cdot V_1(l_1 + 2r)$$

$$\Delta\Gamma = 2n \cdot V_1 \cdot r \tag{14}$$

From the above equation it can be seen that without turbulizators (n = 0) vortex circulation at the output does not change ($\Delta\Gamma = 0$) and for all n > 0 there will be an increase in head due to the vortex circulation at the output increases.

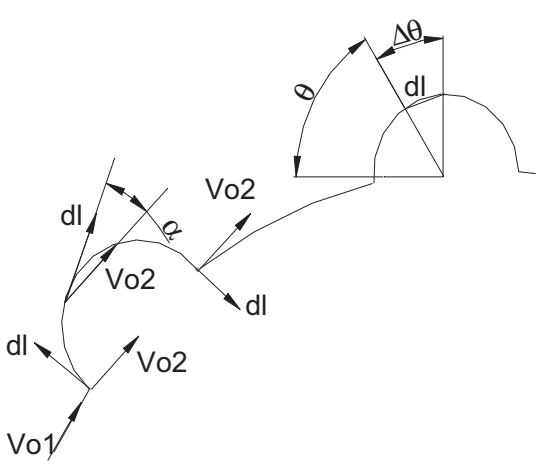

Fig. 12 Velocity without turbulizers.

8. Conclusions

Experiments show that the creation of vortex circulation in the impeller outlet by placing turbulizers of different cross section shapes can satisfactorily be an effective way to increase the head and efficiency of the disc pump, being the square cross section of greater influence. In this case, the creation of vortex circulation in the output impeller contributes to the increase the energy transferred toward the fluid from the impeller.

The number of turbulizers (n), the step or separation between them (l1), and its radius (r) influence on the vortex circulation at the impeller outlet (Γ_2), and therefore influence on the disc pump head (H).

The vortex circulation in the impeller outlet leads to increase head and efficiency of the disc pump up to a certain optimum amount of turbulizers. An increase in the number of turbulizers exceeding its optimum value leads to decrease outflow area below the critical area which is associated with the number of turbulizers that achieves the best results. Therefore, each impeller with a different diameter has an optimal amount of turbulizators that achieves the best head and efficiency of the disc pump.

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