PERFORMANCE ANALYSIS ON A TESLA BLADED DISC PUMP

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ABSTRACT

Tesla Bladed Disc (TBD) pumps are specifically designed to transport all kind of multiphase viscous mixtures and crude materials. Impellers usually have pure radial straight blades fixed on two parallel disks separated by an open clearance gap. The paper presents an analysis based on overall experimental performances on a TDB prototype specifically designed and tested in water for several rotational speeds. The need for careful investigation on power balance terms is highlighted. In addition, CFD results are one of the possible ways to get a better understanding on flow features that are developing in these pumps. First set of CFD results detects local inhomogeneities inside the impeller depending on the flow rates. This contribution is a starting point for further investigation about the ability of such a pump to drive multiphase flow mixtures and more specifically inlet gas-liquid two-phase conditions with an expectation of delayed pump shut-off conditions compared with conventional designs.

KEYWORDS

EXPERIMENTS, TESLA TYPE PUMP, RADIAL BLADE, PERFORMANCE

NOMENCLATURE

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>b</td>
<td>impeller channel width (m). b=2h+cl</td>
<td>-</td>
</tr>
<tr>
<td>cl</td>
<td>inter blade clearance</td>
<td>-</td>
</tr>
<tr>
<td>g</td>
<td>acceleration (m²/s)</td>
<td>-</td>
</tr>
<tr>
<td>h</td>
<td>blade height (m)</td>
<td>-</td>
</tr>
<tr>
<td>H</td>
<td>total head (m)</td>
<td>-</td>
</tr>
<tr>
<td>N</td>
<td>rotational speed (rev/min-rpm)</td>
<td>-</td>
</tr>
<tr>
<td>P</td>
<td>shaft power (W)</td>
<td>-</td>
</tr>
<tr>
<td>Q</td>
<td>volume flow rate (m³/s)</td>
<td>-</td>
</tr>
<tr>
<td>R</td>
<td>Radius (m)</td>
<td>-</td>
</tr>
<tr>
<td>U</td>
<td>impeller tip speed (m/s)</td>
<td>-</td>
</tr>
<tr>
<td>V</td>
<td>absolute velocity (m/s)</td>
<td>-</td>
</tr>
<tr>
<td>Z</td>
<td>number of impeller blades (-)</td>
<td>-</td>
</tr>
<tr>
<td>β’</td>
<td>blade angle (°), from tangential direction</td>
<td>-</td>
</tr>
<tr>
<td>η</td>
<td>hydraulic efficiency</td>
<td>-</td>
</tr>
<tr>
<td>ω</td>
<td>angular velocity (rad/s)</td>
<td>-</td>
</tr>
<tr>
<td>ρ</td>
<td>fluid density (kg/m³)</td>
<td>-</td>
</tr>
<tr>
<td>φ</td>
<td>flow coefficient (-)</td>
<td>-</td>
</tr>
<tr>
<td>φ*</td>
<td>flow coefficient ratio= φ/φD</td>
<td>-</td>
</tr>
<tr>
<td>ψ</td>
<td>head coefficient (-)</td>
<td>-</td>
</tr>
<tr>
<td>ψth</td>
<td>theoretical head coefficient, (-)</td>
<td>-</td>
</tr>
<tr>
<td>Ψth</td>
<td></td>
<td>-</td>
</tr>
</tbody>
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specific speed $= \omega Q^{0.5}/(gH)^{0.75}$

Specific radius $= R^2 (gH)^{0.25}/Q^{0.5}$

**Subscripts**

1. impeller inlet section
2. impeller outlet section
3. design condition

**INTRODUCTION**

When pump users are facing hard to pump materials because of multiphase flow conditions, high viscosity fluids and/or non-homogeneous slurry mixtures, specific mechanically robust design concepts must be developed compared with classical ones. The use of low specific pump category usually called “drag pumps” comes from a simple first design invented around 1850’s in the US (see figure 1a). This pump operates solely on the boundary layer viscous drag principle by means of hole arrangement around a rotating cylinder.

Nikolas Tesla (1913) develops another kind of machine, which enables to use high viscosity fluids between two small spacing co-rotating discs working as a turbine. This kind of device can generally pump a small amount of flow rate with small head coefficient compared with classical rotor-dynamic centrifugal pumps. By adding several co-rotating discs, as shown in figure 1b, the so-called “Disc-pack” design, patented in 1988 by Max I. G, one can only increase the flow rate, but still head values remain low.

When adding several small blades on each co-rotating disc and maintaining an open gap in between, it is possible to reach higher flow rates and head values. An example of such pump is given in figure 2, from “Discflo™” Corporation patent. This arrangement can be considered equivalent to a double face to face open centrifugal pump impeller that rotates at the same angular speed with a large gap in between. One can also find another kind of impeller shape proposed by Gülich (2010), called “double acting overhung impeller” with 20 radial blades on each side that can deliver higher outlet pressure.

Tesla disc pumps have been already studied in the past years. However, a few results can be found on Blades Disc or Tesla Bladed Disc (TBD) pumps, but generally not well documented. Their unusual quite crude design, compared with classical impeller one, however reveals a renewed interest in relation with safety and sustainable development process. It securely allows to transport a large variety of multi-phase sludge materials with basically low efficiency values. The combination of shear stress force acting on the rotating disks and pressure force on small radial blades heights can provide sufficient power to the fluid, but never reach conventional impeller pump design performances.

Maximum global efficiency can reach 0.5 for good technological designs and is more generally closed to 0.4 – 0.45 depending on the inter-blade gap values as already checked by Gao (2009), Zhou (2012), Zhang (2015), Je and Kim (2019). In addition, most of experimental and numerical results that are available in open literature, do not correspond to the pump optimum conditions because of strong incidence angles at impeller inlet and non-optimum volute design. In addition, available numerical results do not include complete pump geometrical parameter information.
Figure 1a. Drag pump.  Figure 1b. Disk-pack pump  Figure 2. Discflo™ pump

Figure 3. Present impeller main dimensions.
PROTOTYPE TEST CASE.

Figure 4. Meridional TBP pump section for:

a) present case (left-dimensions are in mm), b) Discflo™ Corp. case (right).

The present TBD prototype is experimentally tested on a specific test bench, which is described by Heng et al. (2020) using water as working fluid. Main impeller geometrical parameters are given on figure 3. Table 1 includes some published work data derived from the impeller geometry from Discflo Corp.™. Last reference of table 1, corresponds to an unshrouded impeller with a tip clearance ratio \( c_l/b_2 = .112 \) equivalent to \( h_2/b_2 = .888 \), from Choi et al. (2006). It has been added because its performances, with a smaller clearance value, looks similar to present TBP design.

Table 1. List of TBD main geometrical pump characteristics from literature.

<table>
<thead>
<tr>
<th>Reference</th>
<th>( R_2 )</th>
<th>( R_1/R_2 )</th>
<th>( N )</th>
<th>( b_2/R_2 )</th>
<th>( h_2/b_2 )</th>
<th>( Z )</th>
<th>( \beta'_2 )</th>
<th>( \beta'_1 )</th>
<th>( \Phi_D )</th>
<th>( \Psi_D )</th>
<th>( \Omega_s )</th>
</tr>
</thead>
<tbody>
<tr>
<td>Yin (2012)</td>
<td>0.16</td>
<td>.40</td>
<td>2900</td>
<td>0.33</td>
<td>0.28</td>
<td>8</td>
<td>90</td>
<td>90</td>
<td>.026</td>
<td>.48</td>
<td>.34</td>
</tr>
<tr>
<td>Je and Kim (2019)</td>
<td>0.11*</td>
<td>.363</td>
<td>1750</td>
<td>0.33</td>
<td>0.25*</td>
<td>8*</td>
<td>90*</td>
<td>90*</td>
<td>.0</td>
<td>.45</td>
<td>.45</td>
</tr>
<tr>
<td>Present study</td>
<td>0.10*</td>
<td>.56</td>
<td>1000/2900</td>
<td>0.16</td>
<td>0.25*</td>
<td>8*</td>
<td>90*</td>
<td>90*</td>
<td>.08</td>
<td>.28</td>
<td>.8</td>
</tr>
<tr>
<td>Zhou (2012)</td>
<td>0.16</td>
<td>.40</td>
<td>2900</td>
<td>0.32</td>
<td>0.25</td>
<td>8</td>
<td>90</td>
<td>90</td>
<td>.04</td>
<td>.38</td>
<td>.32</td>
</tr>
<tr>
<td>Choi et al. (2006)</td>
<td>0.06</td>
<td>.25</td>
<td>700</td>
<td>0.15</td>
<td>.888</td>
<td>6</td>
<td>90</td>
<td>90</td>
<td>.04</td>
<td>.40</td>
<td>.2</td>
</tr>
</tbody>
</table>

* These numbers correspond to the same geometrical design parameters used in the present pump.

Figures 4a and 4b respectively show meridional cutting planes of the present TBP and an industrial set up from Discflo™ Corporation for comparison. Present case has been adapted and inserted inside an existing volute casing which allows impeller geometrical variations like outlet radius and/or width ratios. Note that the present impeller width is twice narrower than all other published cases. Other main differences concern both hub and shroud disc chambers that are quite smaller for Discflo™ Corp case and the corresponding volute design section which have a squared.
shape. Discflo™ technology is much more improved than present case study; consequently, overall pump efficiency is slightly higher compared with the present case because of a better control on leakage recirculating flows.

Figure 5. Cordier diagram showing the different specific speed and radius values from literature.

Since specific speed only provides a rough indication on the pump geometry configuration, one can have the same value for different designs, depending on each impeller outlet widths and inter-blade gaps. The values of each specific speed and specific radius are respectively given in figure 5, on the so-called Cordier’s diagram.

EXPERIMENTAL PUMP OVERALL PERFORMANCE RESULTS AND DISCUSSION

Experimental data reduction results lead to the performance curves that are given in figures 6 and 7, respectively for head coefficients and pump efficiency versus flow coefficients, for several rotational speeds. (The pump efficiency includes hydraulic and volumetric losses). Details on devices used to measure pressure level, volume flow rate and shaft torque with uncertainties and error
evaluation can be seen in Heng et al. (2020). Error bars correspond to uncertainty evaluation within a range of ±5%. This is mainly due to pressure instabilities that have been observed at pump inlet. Unsteady pressures sensors have not been yet used for the present study. Maximum efficiency value reaches 0.4 when flow coefficient is 0.08 (figure 7). These efficiency values differ from previous results from Heng et al (2020) who obtained global efficiency from torque measurements with a maximum value of 0.35. The present hydraulic efficiency is obtained using an evaluation of the disk friction power calculated using the relation given below, according to Gülich (2010).

These efficiency values differ from previous results from Heng et al (2020) obtained from torque measurements with a maximum efficiency value of 0.35. They are presently corrected by an evaluation of the disk friction power calculated using the relation given below, according to Gülich (2010).

\[
P_{\text{disk}} = k_{\text{disk}} \cdot \rho \cdot \omega^3 R_2^5 \left(1 - \left(\frac{R_1}{R_2}\right)^5\right)
\]

A value of \(k_{\text{disk}} = 0.005\) was initially applied to the present case for a Reynolds number (based on outlet radius) equal to 10^6 and a surface roughness coefficient is equal to 0.004. \(k_{\text{disk}}\) has been also corrected because of a high value of the axial gap between hub disk and volute casing. A value of \(k_{\text{disk}} = 0.01\) is finally retained including leakage effect correction (Gülich (2010)).

Looking at figure 6 and 7 which have been reported in previous study (Heng et al, 2020), one can observed that the well-known similarity law is not fulfill for all rotational speeds. For 2900 rpm, for high flow rate values, cavitation is detected due to low inlet static pressure detection (the right side last 4 experimental points on figure 5 are under cavitation conditions). For 1000 rpm, insufficient experimental accuracy may explain the pressure coefficient differences. However, they always exhibit higher values for all given flow rates. Further investigation is needed here maybe because: 1) part of the total work can also be done by viscous forces that develop on both external disk parts in relation with Reynolds number, 2) inlet blockage and recirculation may occur at low flow rates, 3) leakages through shroud gap at impeller inlet and equilibrium holes (see location on figure 3, close to the hub root) are not yet well evaluated.

By eliminating disk friction losses, as explained before, make results consistent with the application of the moment of momentum equation that leads to the Euler relation and the related theoretical head coefficient shown on figure 8. Note that one approximation is remaining, since the real flow rate, going through the impeller, can be higher than the delivered flow rate given by the
pump due to leakage effects. Comparisons and discussions with CFD results given in figure 9 will be presented in next section.

The extrapolated straight line, built on theoretical head coefficients higher than \( \phi^* = 0.8 \), using values obtained at coefficients reaches to a value of 0.9 for zero pump flow rate. Such curve is intentionally used here with regards to usual Euler curve obtained for impeller conventional designs, for which the value of 1 is normally reached, under specific assumptions (constant impeller outlet relative flow angle, no impeller local recirculation and no inlet swirl).

For the present case, the limiting case of a 90 degrees relative flow angle (for infinite number of blades) at impeller exit plane, cannot be used because of the existing gap between the two parallel disks. Consequently, it is up to now impossible to calculate, a slip coefficient value at impeller outlet section, using existing correlations which are basically obtained using a one-dimensional approach and mean flow characteristics assumption.

![Figure 8. Theoretical head obtained from experimental results. (Results inside the red area are influenced by cavitation development).](image)

Present experimental results are to be compared to those presented by Hasinger et al. (1963), who develop an analytical approach according to Breiter and Pohlhausen (1962), for laminar flow. It was found that, applying the Hagen-Poiseuille equation to the radial flow between two disks, the head loss varies like \( \Omega^{-1} \) and the outlet slip coefficient varies like \( \mu^{-1} \). This may explain why performance coefficient values do not obey to rotating machinery similarity laws and depend on rotational speed through Reynolds number effects (see figure 5). Note that for the case of pure shear-force pump (without blades), high head coefficient cannot be achieved according to Hasinger et al. (1963) paper, when a width ratio \( b_2/R_2 = 0.5 \) is applied.

Due to the impeller geometry, flow inside the rotating parts probably results from combination of a pure shear-force pump and a rotodynamic one with strong interactions due to gap leakage. Each effect cannot be isolated, and the direct consequence is that any simplified approach will fail without deep local flow investigations.

**CFD RESULTS**

URANS calculations are performed using the \( k-SST \) turbulence model closure assuming turbulent flow conditions. Mesh sensitivity analysis is performed for a low flow rate coefficient \( \phi^* = 0.2 \) which includes strong inlet reverse flow features. According to the results given on figure 9, more than 8 million unstructured elements are used to reach both head and efficiency stable results within a difference of \( \pm 0.6\% \) and \( \pm 0.8\% \) respectively. Whole pump geometry including inlet tube (more than
10 inlet tube diameters upstream), impeller and volute, gaps, leakage clearances plus 8 pressure balancing holes drilled on the rear disk hub.

![Figure 9. Mesh independence analysis.](image)

Figure 9. Mesh independence analysis.

![Figure 10. Head characteristic curves from CFD.](image)

Figure 10. Head characteristic curves from CFD.

![Figure 11. TBP Efficiency curves from CFD.](image)

Figure 11. TBP Efficiency curves from CFD.

The overall results concerning head coefficient and efficiency extracted from CDF, are shown on figures 9 and 10 respectively for 3 rotational speeds. Results for rotational speed of 2900 rpm are not shown due to convergence stability problems at high flow rates. CFD results look in good agreements compared with experimental ones. Using the similarity law rules, results show a slight decrease of head coefficient and efficiency values for the lowest rotational speed that can be attributed to the turbulent model that is applied for these calculations.

The theoretical head coefficient evolutions slightly differ from experimental ones (see comparison between figures 8 and 12). Theoretical curves obtained with CFD are slightly shifted towards higher flow coefficient value, the consequence of which leads to a different slope value compared with the experimental one. This shift can be explained by recirculating flow coming from equilibrium holes responsible for an increase of the real flow rate inside the impeller, responsible for the so-called volumetric efficiency. This point still needs more attention and investigation on leakage flow evaluation and on local mesh density and closure on flow modelling.

An illustration of instantaneous streamline patterns is given in figure 13, for the lowest flow rate $\phi^*=0.2$. Strong inlet reverse flow is found and extends upstream in the inlet tube. For $\phi^*=0.5$ (not shown here), no inlet recirculation is found. These qualitative results will undergo more quantitative...
analysis in future work, focusing on the capability to use a one-dimensional approach for engineering performance prediction.

![Figure 12. CFD results on theoretical head versus flow coefficient.](image)

Figure 12. CFD results on theoretical head versus flow coefficient.

![Figure 13. Streamline patterns inside the inlet tube, off design condition $\phi^*=0.2$. White color arrow indicates mean flow inlet direction.](image)

Figure 13. Streamline patterns inside the inlet tube, off design condition $\phi^*=0.2$. White color arrow indicates mean flow inlet direction.

**CONCLUSIONS**

1. A preliminary experimental study has been performed on a centrifugal specific Tesla bladed disc pump design, using water as working fluid. These results show that acceptable performances can be achieved with such a crude impeller design.

2. Careful data reduction process must be performed concerning external mechanical and leakage losses for further flow passage loss prediction. Otherwise, wrong interpretation on existing published results can be made.

3. CFD results performed using URANS approach give a good evaluation on overall performance within 3% discrepancy compared with experimental results.

4. Additional physical analysis is necessary for a better understanding of local flow pattern and related losses levels to propose engineering tools based one dimensional approach for slip velocity evaluation at impeller outlet and loss prediction inside the impeller and the volute.
5. Off-design conditions are always expected due to the radial blade shape. More detailed experimental work must be performed between the blade open space and in the upstream tube section to check inlet loss accumulation related with swirling flows and their consequences in case of inlet two-phase flow conditions which is the final aim of this work.

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