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VALIDATION OF TURBODIL FLOW RATE PREDICTIONS

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ABSTRACT

Feed dilution is frequently applied to thickener feedwells to improve the flocculation process. Various methods are employed for either natural or forced dilution, including a Turbodil, which is essentially a low-head pump. There is no equivalent commercially available pump, and typical alternatives (e.g. axial flow pump, centrifugal pump, displacement pump) are inefficient in the required head range.

The Turbodil has recently been redesigned with an extensive CFD optimisation program. The CFD itself was relatively straightforward, being mostly single-phase, steady (RANS) turbulent, isothermal and incompressible. As such, it was readily handled with a commercial solver (Fluent), using either multiple reference frames or transient sliding grid for the rotating impeller, with no modification to the standard code.

The emphasis of this paper is on the pilot-scale validation. A 20 m^3 rectangular test tank was constructed with a central divider, giving two equal sized compartments. Water was pumped from one side to the other with various configurations of Turbodil, and allowed to return via an adjustable gate that controlled the level (head) difference. Impeller rpm, power draw, flow rate and head were adjusted or measured to give a set of flow curves for validation of the CFD predictions. Following the CFD validation the new Turbodil design has been "productised" into a range of sizes giving flows over the range 100-10,000 m^3/h.

INTRODUCTION

Gravity sedimentation is widely used in the mineral processing and water treatment industries to perform solid-liquid separation. Flocculant is normally added to bind the fine particles together into aggregates, which increases the settling rate and overflow clarity. The flocculation process is complex, but in broad terms at least, reasonably well understood, with the following basic steps (Healy 1961, Gregory 1989):

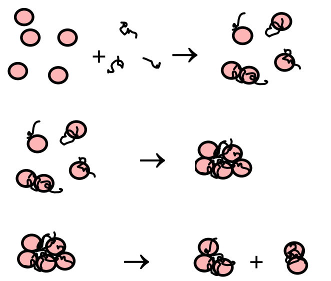

Figure 1: Flocculation process

Flocculation is essentially a mixing problem, but must be done relatively gently to avoid breaking up the fragile aggregates. Mixing, turbulence, fluid shear and the mixing time are critical to good flocculation. Fortunately they can also be controlled and significant engineering development has been undertaken in the last few years to improve thickener feedwell design (Triglavcanin 2008, Heath & Triglavcanin 2010, Nguyen et al. 2006, Nguyen et al. 2012, Echeverri et al. 2012, Krassnokutski 2012).

Aggregate formation follows fractal geometry (Meakin 1988, Horwatt et al. 1992, Heath et al. 2006a), meaning the larger the aggregate the more porous or fluffy it is. It also takes up more space volumetrically in the slurry, and concentrated slurries are typically diluted to improve the flocculation process. There is an optimal solids concentration for good flocculation. Too low and the particles are too far apart, resulting in too few collisions (second order kinetics) and poor flocculation. Increasing the agitation rate increases the collision rate, but also the breakage rate (Heath et al 2006b, 2006c). Consequently, over-diluted feeds can lead to slow flocculation and dirty overflow liquors, a situation not helped by an increased rise rate in the thickener's clarification zone due to the increased liquor flow. In some cases underflow is recycled back to the feedwell to increase the solids concentration.

If the solids concentration is above the optimal value then it becomes increasingly difficult to mix the flocculant successfully without localised overdosing. Also, due to the fractal nature (i.e. increasingly porous, or fluffy, with size) of aggregates they rapidly run out of space and approach the gel point where they completely fill the available space. While it may seem somewhat counterintuitive to begin the gravity thickening process by diluting the feed, the improvement in flocculation is sufficiently dramatic that the overall solid settling flux (unit throughput) are improved. The optimal feed dilution is normally determined by testwork.

There are various methods for dilution in operating thickeners, e.g. feed eductors (Krassnokutski 2012) or dilution ports (Triglavcanin 2008). Overflow liquor can also be pumped into the feedwell or feed pipe. This could be done with a conventional centrifugal pump, however this is excessive in both capital and operating costs because large volumes of dilution are required, but only a trivial pumping head is required. Conventional centrifugal pumps are significantly over-engineered for such low heads, and are also inefficient in that head range. Alternatively a Turbodil can be used, which is essentially a purpose built high-volume low-head pump. Previously turbodils were sourced from a sub-supplier and used a mixing type impeller in a draft tube. Turbodils have recently been redesigned and optimised with a combination of CFD and pilot-scale validation, with the design taken in-house at Outotoec.

CFD MODEL SET-UP

The CFD simulations were all run in ANSYS-Fluent using standard features available in the GUI, with no additional modifications or user coding. Note the emphasis of this paper is on the CFD validation and product development, there is no CFD development per se.

The majority of the simulations run as steady-state (RANS) Eulerian, single phase, Realisable k-ε turbulence model, MRF (Multiple Reference Frames) for the impeller rotation, and mixed hex/polyhedral meshes of ~700k solver nodes. More computationally expensive sliding grid simulations were used in a limited number of simulations as indicated in the text as a more rigorous comparison to MRF. Finer grids were used as indicated to check for mesh independence (minor effect), and various alternate turbulence models were also tried for comparison (again minor effect). Some multiphase (air + liquor) simulations were also run with the modified existing Turbodil to simulate vortexing and possible air entrainment.

CFD RESULTS AND DISCUSSION

Turbodil pumps have been supplied since 1997 for feedwell dilution. The original design consisted of a vertical draft tube with a mixing tank type impeller similar to an A310 to draw the liquor upwards and then along a horizontal leg into the feedwell or feed pipe, similar to Figure 2. The draft tube was surrounded by a submerged tub that ensured the draft tube was fed with liquor from the surface, avoiding the potential to drag the settled solids from the top of the settled mud bed. Generally the unit performed as expected, however vortexing was sometimes observed in the tube. Also, the impellers were supplied by a sub-supplier and the flow curves were not verified but were taken to be based on impeller flow calculations for similar impellers in mixing applications, i.e. where there was no pumping head.

Since a Turbodil is essentially a low-head pump, this project began with a literature review and search for similar commercially available pumps. It wasn't obvious at the outset that this was a problem that warranted R&D, let alone CFD. However, very little information could be found, with no commercial pumps available to pump large volumes (100-10,000 m^3/h) against very low heads (<0.5 m). The closest are axial flow pumps for irrigation and process pumping. These were similar to the existing Turbodil arrangement but with much wider impeller bladed (so called "high density" impeller), but efficient in the 2-10 m head range. Conventional centrifugal pumps are designed for higher heads (~10-50 m) and are consequently inefficient and over engineered for low-head applications.

A lack of prior art or commercial products being established a CFD optimisation process was undertaken on two separate designs:

1. Axial flow design as per the existing Turbodil.
2. Radial flow design based on the Outotec DOP impeller used in solvent extraction equipment. A volute shaped housing (e.g. Figure 3) was added around the impeller to make it into a pump, similar in design to a commercial radial flow air blower.

The design process for each was based on a flow of ~400 m^3/h, large enough to be full-scale, but small enough to be amenable for testing with a test-rig.

Design optimisation was via an intuitive route of changing the various possible design parameters and chasing up the pumping efficiency. Optimisation algorithms like SIMPLEX or Newton's method weren't used in this case, except in spirit, due to the large number of variables and long run times. The optimisation process was taken to be complete when there were no more obvious options to try and tweaking the previous ones didn't make any significant improvements.

Over 100 simulations were run on each of the two configurations. Using the previous run as an initial guess typically lead to a converged result in a couple of hours using multiple processors. Convergence was generally straightforward, as would be expected of a relatively simple incompressible single-phase RANS flow problem. The various simulations run towards the optimised designs won't be discussed here in detail, suffice to say that the impeller blade angles, number, depth, width etc were varied, along with the housing geometry. A formal mathematical objective function wasn't used here, but the important criteria were:

- Maximum pumping efficiency, i.e. conversion of shaft power to useful pumping work.
- Maximum flow rate. Primarily to keep the unit size and cost down. A 5000 m^3/h pump is physically fairly big, and must be accommodated on the thickener bridge.
- Flow independence from head. The head that the Turbodil has to pump against is not easily calculated in advance, and depends on the transition box and feedwell configuration and process variables like the feed flow rate and solids concentration. Ideally the flow rate would be dependent only on the rotation rate (i.e. as per a displacement pump) and unaffected by the head so that the unit could be installed with a given gearbox ratio or VSD frequency and produce a known flow rate.

- Width of efficient flow range. The intention from the outset was to develop a product range of different standard size Turbodils. Having them pump efficiently over a wider flow rate range would result in fewer sizes, and also allow more scope for tuning once installed.

- Ease and cost of manufacture, installation and repair.

The final optimal designs are shown in Figure 2 through Figure 5. These won't be described in detail here although it is noted that both designs benefited greatly from having an x-shaped baffle placed immediately below (upstream) of the impeller. This prevented vortexing which was prone to occurring at higher heads (~0.2 m and above). The axial flow impeller in particular was prone to this, and it resulted in a "block" of fluid rotating with the impeller and no flow, rather than the impeller cutting through the fluid. Placing the baffles above the impeller (downstream) was virtually ineffective, although the square shaped draft tube did proved some additional baffling.

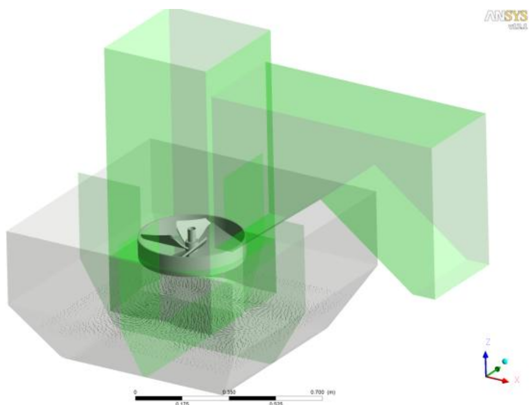

Figure 2: Optimised axial flow configuration.

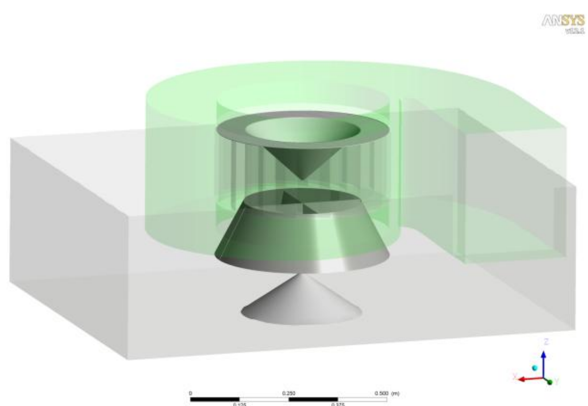

Figure 3: Optimised radial flow configuration.

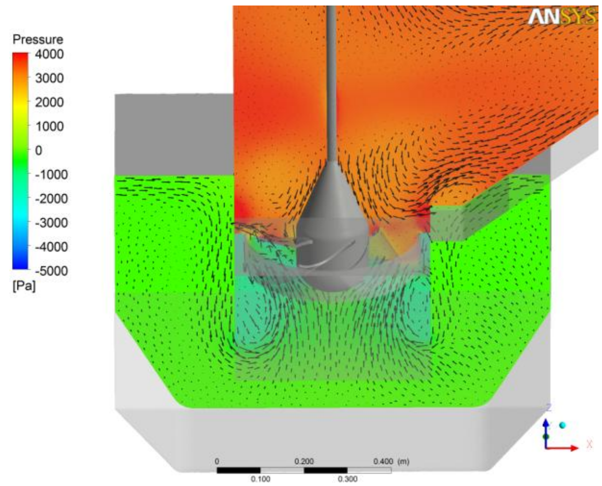

Figure 4: Flow and pressure fields of the axial flow option.

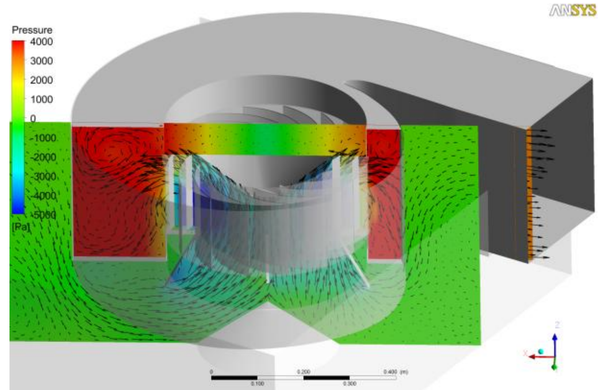

Figure 5: Flow and pressure fields of the radial flow option.

Pilot-Scale Test Rig

A test tank was built from a modified shipping container as per and Figure 6 and Figure 7. This was a convenient size (20 m^3) and shape, and the ribbed sides provided sufficient bending strength to resist the water pressure once the top had been braced with angle iron and flat bar strips. A divider plate was welded in the centre, giving two equal sized compartments. The flow was pumped from one compartment to the other, and returned via an adjustable gate. The gate was adjusted to give various steady-state level differences between the compartments in the range 0-0.3 m.

Figure 6: Test tank built from modified shipping container.

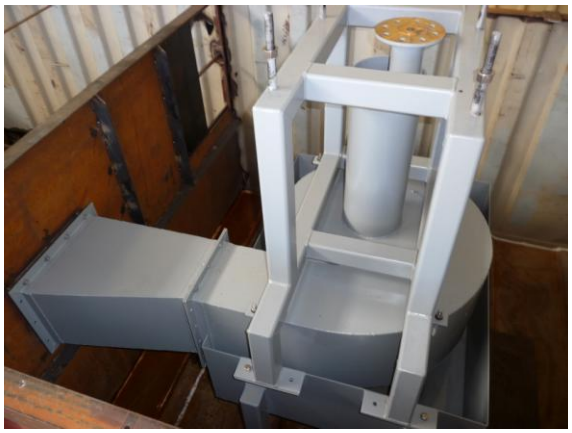

Figure 7: View of the inside of the test tank. The Turbodil pumps from one compartment to the other, returning via the adjustable gate in the central divider.

Figure 8 and Figure 9 shows the tank in operation at two different gate adjustments, giving either 0.3 m or ~ 0 m head difference. The impellers were run through a range of speeds to establish the flow curves, with the flow measured by a hand held flow probe placed in the Turbodil outlet into the far compartment.

Figure 8: Radial flow impeller at 250 rpm with 0.3 m head.

Figure 9: Radial flow impeller at 250 rpm ~0 m head.

Comparison of CFD and Measured Results

The flow and power draw results from CFD and measurement are shown below in Figure 10 to Figure 13. Clearly the flow results (Figure 10 and Figure 12) are remarkably close, especially the low head figures. This gives the confidence to use the CFD to develop a range of (mostly) larger size units for commercial use. The power draw numbers are obviously different (Figure 11 and Figure 13), however the CFD predictions are for the shaft power to turn the impeller, whereas the measurements are from the VSD used to drive the electric motor. Since there will be losses through the drive system, especially the chain drive, the results appear plausible.

In terms of the results themselves the radial flow design is more efficient and is less affected by head, and hence was the chosen design. Subsequently, but not shown here, the radial flow design was translated into 10 different sizes covering the range ~100-10,000 m^3/h. Some overlap in flow was allowed between the designs to help the selection process. Several units have subsequently been installed on site and are operating as expected, and the on-site flow measurements will be presented in a future publication.

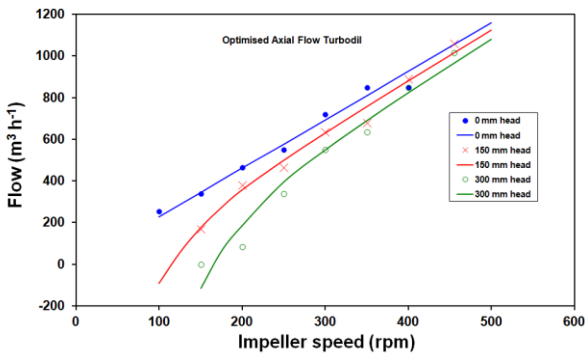

Figure 10: Flow vs. impeller speed for the axial flow Turbodil. Lines are the CFD predictions, dots are the measured values.

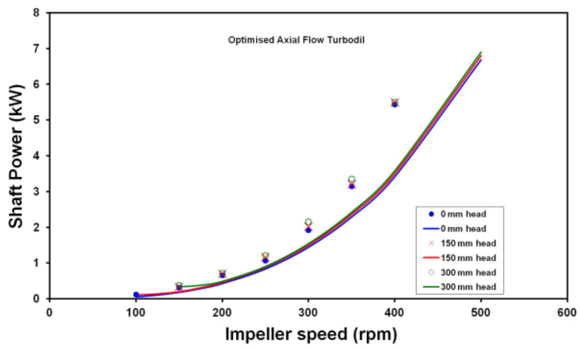

Figure 11: Power vs. impeller speed for the axial flow Turbodil. Lines are the CFD predictions, dots are the measured values. Note CFD is just the power to turn the impeller, whereas the measured values include losses from the motor and chain drive.

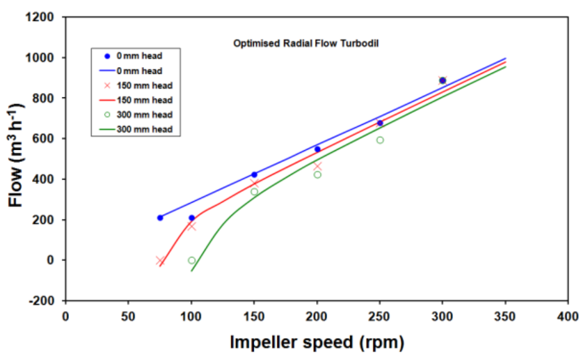

Figure 12: Flow vs. impeller speed for the radial Turbodil. Lines are the CFD predictions, dots are the measured values.

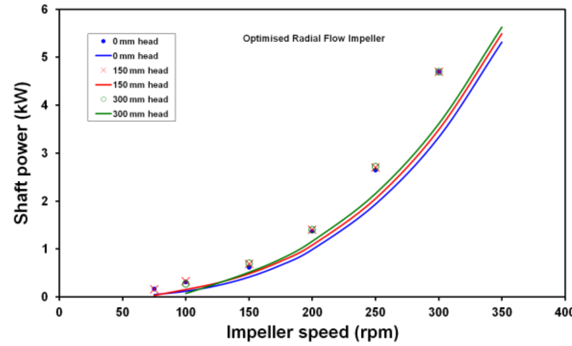

Figure 13: Power vs. impeller speed for the radial flow Turbodil. Lines are the CFD predictions, dots are the measured values.

Application of CFD Results to Mechanical Design

In addition to the flow and power predictions the CFD results were also used to predict the side loads on the impeller in the radial flow design. This is important to the mechanical design of the impeller drive shaft which is moderately long and only supported by bearings at the top (above the liquor/slurry surface). Differential pressure through the casing was resolved to determine the net pressure force imposed on the impeller. The net force acts in a constant radial direction and applies a bending load to the shaft. Due to the constant nature of the load, the shaft experiences fully reversed bending as it rotates through 360 deg. The fatigue strength of the steel shaft was derived from the S-N curve for the material. The shaft was sized such that the combined reversed bending stress was less than the stress range dictated by the fatigue strength of the shaft.

CONCLUSIONS

A combination of CFD and pilot-scale testing has been used to develop an optimised range of new Turbodils based on a radial flow impeller. The CFD itself was not exotic, being simply single phase, incompressible, RANS. Multiple reference frames were used for the sliding grid, and solved in Fluent. Convergence was essentially straightforward, and required no exotic solver settings. The CFD predictions were in good agreement with the subsequent measured results, and the model has now been used to develop an improved commercial range of Turbodil pumps.

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