20th Australasian Fluid Mechanics Conference
Perth, Australia
5-8 December 2016

Effects of a new-design diffuser and a tail piece on an axial-flow pump's performance

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Abstract

Measurements have been conducted to gauge the effects of a new-design diffuser and an additional tail-piece on the performance of an industry-sized axial-flow pump unit. The new-design diffuser was seen in a previous study to reduce the swirling motion that would eventually dissipate as lost energy and was still present in the flow as it exits from the standard pump-unit that was originally fitted with a conventional conical diffuser. The additional tail-piece, which is conical with an apex angle of 20° is fitted to the new diffuser's exit to allow for a gradual change in the flow area from the diffuser's exit to the discharge pipe, instead of a sudden expansion. Impellers with 5 vanes and 8 vanes have been used. With the 8-vane impeller, an improvement in performance has been obtained, compared with the standard pump, when both the new diffuser and the additional tail-piece are used together. With the 5-vane impeller, however, the standard pump performs better. CFD (Computational Fluid Dynamics) analysis using ANSYS CFX has also been conducted; and this also shows the benefits of the additional tail-piece when the 8-vane impeller is used.

Introduction

During the 1990's a new type of diffuser was designed for an Ornel axial-flow pump, for improving its performance by reducing the fluid swirl which is still present in the flow as it exits from the standard pump-unit that was originally fitted with a conventional stator followed by conical diffuser. This residual swirl would eventually dissipate as lost energy and hence reducing pump efficiency.

Previous laboratory research showed there is an increase in pressure along the new diffuser when compared to the conventional stator-conical-diffuser, hence a possible reduction in the fluid swirl. However the overall pump performance was less than the original equipment manufacturer (OEM) figure. One main reason was believed to be due to a sudden enlargement at the outlet of the new diffuser causing much wasteful re-circulation [1] (see Figure 1)

To better determine the source of degradation of the overall pump-performance and possible improvements which could be made to the new diffuser, dimensional investigation, computational fluid dynamics (CFD) analysis and laboratory experimentation were conducted.

CFD modelling was used to help predict the flow patterns and turbulence through the axial-flow pump and also as part of the design process for a tail-piece component. The tail piece was to be installed at the outlet of the new diffuser to mitigate the sudden enlargement's detrimental effects.

Experimental laboratory testing was then performed using the test rig located at the University of Technology Sydney. By using new pump parts combined with the new diffuser and tail-piece addition, performance of the pump unit was determined.

Analysis and Design

The pump used for this work is an Ornel 300AX axial-flow pump. The original configuration is a single stage with a standard stator-conical-diffuser arrangement. The new diffuser replaces the standard arrangement and was designed to reduce the residual swirl still present in the flow, as found in a previous study [1]. However the new design introduced a sudden enlargement at the outlet of the diffuser which creates flow separation and a large low-pressure recirculation region resulting in pump-efficiency reduction, as the CFD analysis shows in Figure 1.

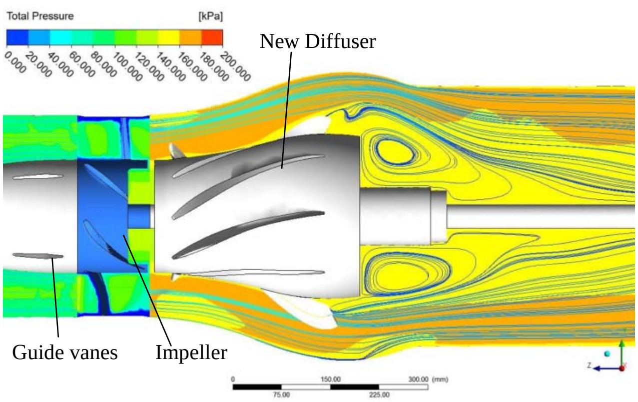

Figure 1. 2D-streamline flow-pattern and total-pressure distribution through pump with the new diffuser. Flow direction is from left to right.

A coordinate measurement machine (CMM) was used to measure the new diffuser by laser scanning the surfaces. From the CMM scanned-image a 3D model was created to determine the parts' dimensions and be used with the CFD analysis. From the model the diffuser-vane inlet-angle α_3 was established at three sections and compared to the impeller's absolute outlet-velocity angle α_2 of both the five-vane and eight-vane impellers. The recommended tolerance between the two angles is ±5° [2]. It was discovered there is a mismatch between the angles for the five-vane impeller, however a good match for the eight-vane impeller; this is shown in Table 1. This suggests that the new diffuser is more suited to high-flow applications with the eight-vane impeller.

To remove the sudden enlargement from the new diffuser outlet and minimise its detrimental effects, a tail piece was designed with the help of CFD analysis. A number of different design shapes were analysed using turbulence models within ANSYS CFX. The turbulence model selected was Shear Stress Transport (SST) which is a variant of k-ω and k-ε models. The k-ε models are the standard for fully turbulent flows but studies conducted by Menter [3] showed that the prediction of boundary layer separation was not accurate, whereas a k-ω model would give a more accurate prediction of fluid turbulence close to a wall. All CFD models were run using the best efficiency point (BEP) from the OEM pump curves. For the five-vane impeller, flow is 318 L/s and 564 L/s for the eight-vane impeller.

Angle	Section		
	AA	BB	CC
5 vane α_2	33°	34°	36°
α_3	55°	59°	63°
5 vane Δ	22°	25°	27°
8 vane α_2	51°	56°	61°
α_3	55°	59°	63°
8 vane Δ	4°	3°	2°

Table 1. Impeller's absolute outlet-velocity angle α_2 for 5-vane and 8-vane impellers, and new-diffuser vane inlet-angle α_3. Δ is the difference between α_3 and α_2. Sections are at radial distance of 174 mm (AA), 235 mm (BB) and 304 mm (CC). All angles are measured relative to the tangential direction.

The CFD analysis shows that a conical tail-piece with apex angle 20° results in the least amount of recirculation close to the diffuser-outlet vane tips and the following area, as shown in Figure 2. Due to design restrictions with possible interference between the tail piece and the line shaft coupling, 20° is the smallest apex angle which could be used.

Comparison of pressure rise between the new-diffuser-only arrangement and the new diffuser combined with the tail-piece addition using a five-vane impeller is shown in Figure 3 and Figure 4. Pressure at seven evenly spaced cross-sectional planes along the diffuser were constructed through the CFD model. It was observed that with the tail-piece addition a greater percentage of the cross-sectional planes has higher pressure. This thus suggests that the gradual reduction of the flow-passage's cross-sectional area thanks to the tail-piece helps to sustain higher pressure for a prolonged distance through the new diffuser. This result was also found with an eight-vane impeller, as shown in Figure 5 and Figure 6.

To manufacture the tail piece a 3D printer was used. This method was selected over casting due to the low manufacturing cost. With casting, finish-machining and pattern production are required, plus the possibility of defects such as porosity, material shrinkage and poor surface finish. Thus 3D printer was deemed a more accurate and cost effective option. In Figure 7 the 3D model of the tail piece and the method of installation are shown. The material used is Acrylonitrile Butadiene Styrene (ABS) which has a high impact resistance and corrosion resistance and is a common material used in the manufacture of clear water piping and fittings.

Laboratory Testing

The pump testing rig is a closed loop system with the axial-flow pump mounted in a horizontal position. The net-positive-suction head (NPSH) for the system is delivered via a booster pump fed from the under-floor tank and manually controlled via a pressure control valve. The arrangement is shown in Figure 8.

For this work six different configurations of the axial-flow pump were assembled and tested at the recommended speed of 1465 rpm as per the OEM pump-curves, and the configurations' performances are compared. The six configurations are shown in Table 2.

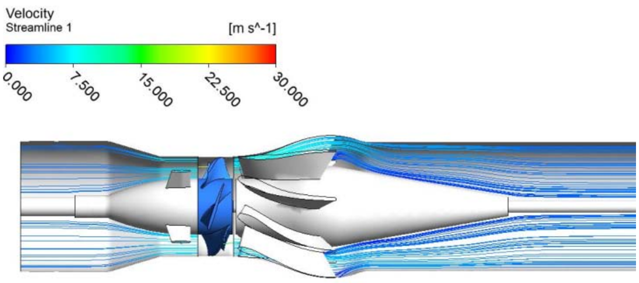

Figure 2. Five-vane-impeller 2D streamlines of new diffuser and tail-piece addition

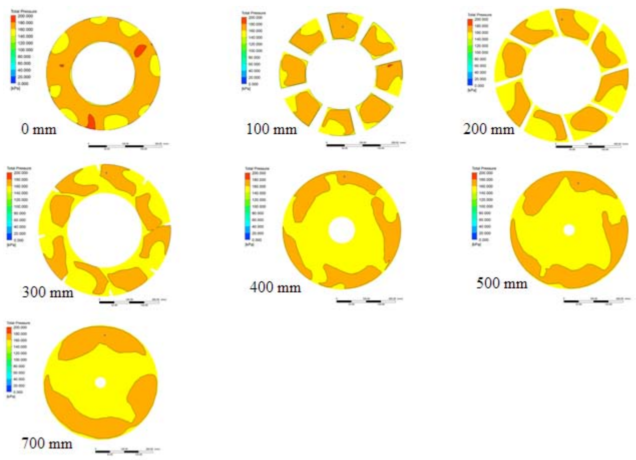

Figure 3. Total pressure on sections through the new diffuser for five vane impeller. Dimensions 0 mm to 700 mm indicate the distance downstream from the diffuser's inlet face.

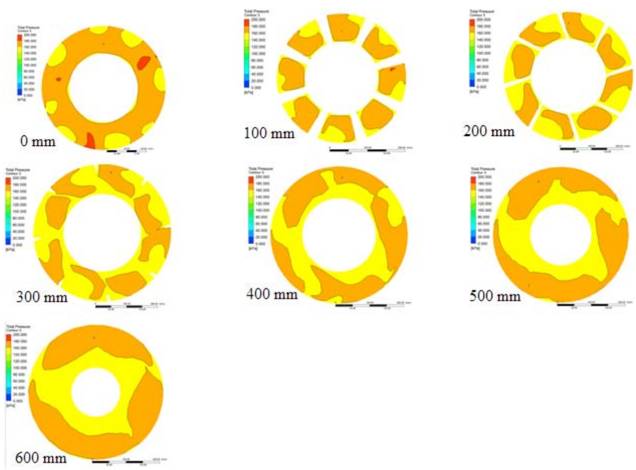

Figure 4. Total pressure on sections through the combined new-diffuser and tail-piece for five vane impeller. Dimensions 0 mm to 600 mm indicate the distance downstream from the diffuser's inlet face.

The pump-test curves for different configurations using the 5-vane impeller are shown in Figure 9. Test results at peak efficiency for the different configurations using the five-vane impeller are shown in Table 3. The table compares results from the base configuration (with standard stator-conical-diffuse arrangement), configuration with new-diffuser only, and configuration wherein both the new diffuser and the tail-piece are used. Figure 9 and Table 3 show that the new-diffuser configuration underperforms when compared to the base configuration with a drop in head of 9.2% but an increase in flow of 3.8%. With the tail-piece addition, efficiency and pump-head curves were slightly reduced compared to the new-diffuser only; however the stall line of the pump was improved (Figure 9). Also, pump head with new-diffuser plus tail-piece addition is still lower than the base test's. The poor performance with the 5-vane impeller is believed to be due to the mismatch of angles mentioned above (Table 1)

On the other hand, testing using the eight-vane impeller gave promising results. Comparison of pump-test curves is shown in Figure10. Results at peak efficiency for the different configurations are shown in Table 4. Compared with the base-test's (with standard stator and conical diffuser), results of new-diffuser-only configuration show a slight increase in peak efficiency (Table 4) but also a reduction in the pump-head curve (Figure 10). On the other hand, with the tail-piece addition to the new diffuser, the pump-head curve is relatively the same as the base-test's (Figure 10), but peak efficiency is clearly increased by about 3.9%. The pump stall line remains about the same.

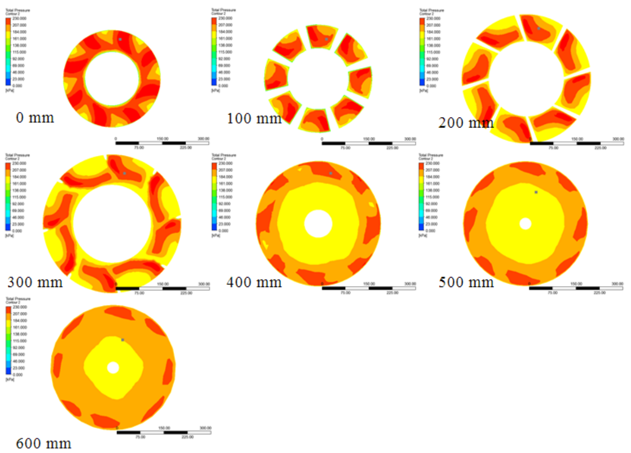

Figure 5. Total pressure on sections through the new diffuser for eight-vane impeller. Dimensions 0 mm to 600 mm indicate the distance downstream from the diffuser's inlet face.

Conclusion

With the 5-vane impeller for medium-flow applications, the new-diffuser configuration underperforms when compared with the standard pump having separate stator and conical diffuser, even with a tail-piece installed. Even though there is an improvement in the pumps operating limit to the new diffuser when the tail piece is added, the original pump is still the best performer. Main reason for the performance degradation is attributed to the mismatch of the new diffuser's vane angles.

However, with the 8-vane impeller for high-flow applications the new diffuser's vane-angles match the impeller's. The combination of new diffuser and a tail piece then greatly improves the pump's performance, with efficiency increased by 3.9%.

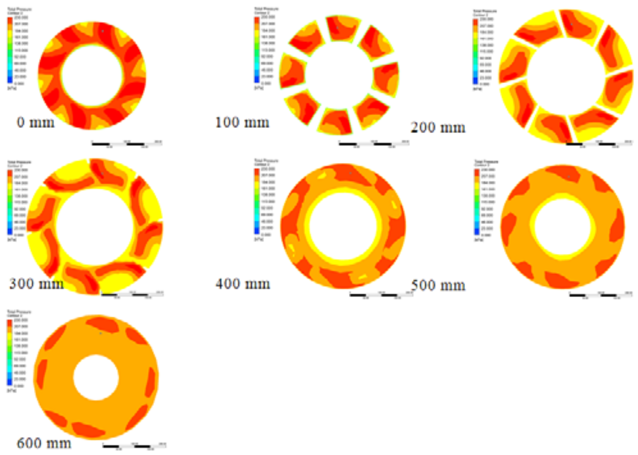

Figure 6. Total pressure on sections through the combined new-diffuser and tail-piece for eight vane impeller. Dimensions 0 mm to 600 mm indicate the distance downstream from the diffuser's inlet face.

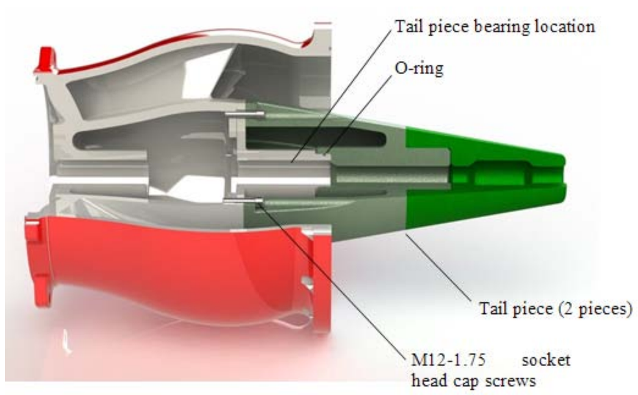

Figure 7. 3D model and installation of printed tail piece. New-diffuser (red) length is 457 mm, total tail-piece's 600 mm. Flow direction is from left to right.

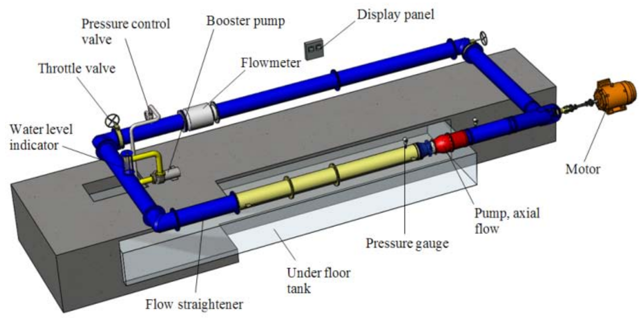

Figure 8. Pump test-rig arrangement (loop) viewed from above; flow direction counter-clockwise. Loop length 9400 mm, width 3000 mm (between pipe centre-lines); pipe's inside diameter 410 mm..

Future Work

Some work which could not be accomplished during this research would be recommended to further enhance the findings.

The purchase of new pump components, specifically an eight-vane impeller and a new medium-flow stator. The existing items are worn and have poor surface finishes adding mechanical losses to the pump.

Laboratory testing using the six-vane impeller. Separate testing with a medium flow and a high flow stator would be beneficial as the six-vane impeller crosses over from the medium to high flow range.

Pump efficiencies of the various configurations using CFD. Unfortunately it was deemed too time consuming to obtain full CFD analysis for this research.

An improved diffuser design for the five-vane-impeller assembly, especially to address the vane-angle-mismatch issue.

Pump Test Configuration		
Test number	Impeller vanes	Pump configuration
1.04	5	Type standard stator + standard diffuser
2.02	8	Type standard stator + standard diffuser
3.02	5	New diffuser design
4.03	8	New diffuser design
5.02	5	New diffuser design + tail piece
6.01	8	New diffuser design + tail piece

Table 2. Pump test configurations

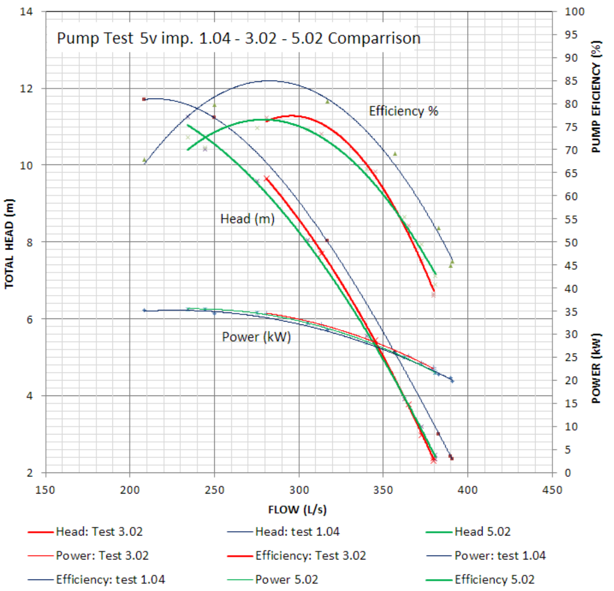

Figure 9. Pump curves for 5-vane impeller. Test numbers 1.04, 3.02 and 5.02 are explained in Table 2.

Test Number	Results					
	Head		Flow		Efficiency	
	H (m)	ΔH	Q L/s	ΔQ	H	Δη
Base (1.04)	9.80	-	284	-	85%	-
New Diff. (3.02)	8.90	-9.2%	295	+3.8%	77.5%	-8.8%
Tail Piece (5.02)	9.30	-5.2%	280	-1.5%	76.5%	-10%

Table 3. Five-vane-impeller pump-test comparison to base test. Test numbers are explained in Table 2. Data are at peak efficiency.

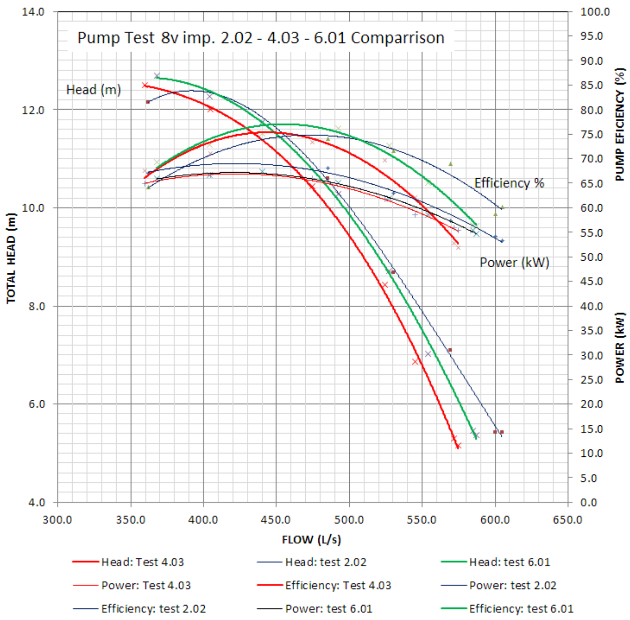

Figure 10. Pump curves for 8-vane impeller. Test numbers 2.02, 4.03 and 6.01 are explained in Table 2.

Test Number	Results					
	Head		Flow		Efficiency	
	H (m)	ΔH	Q L/s	ΔQ	H	Δη
Base (2.02)	10.70	-	475	-	74%	-
New Diff. (4.03)	11.30	+5.4%	445	-6.3%	75%	+1.4%
Tail Piece (6.01)	11.35	+5.8%	460	-3.2%	77%	+3.9%

Table 4. Eight-vane-impeller pump-test comparison to base test. Test numbers are explained in Table 2. Data are at peak efficiency.

Acknowledgements

Florin Voicu (Industrial Engineer, Weir Minerals), Dr. Luis Moscoso (Principal Engineer, Weir Minerals), Tony Nelson (Engineer), Nick Goodall (Regional Manager, LEAP Australia), Peter Tawadros (Mechanical Lab Manager, UTS) and Peter Brown (Senior Engineer, UTS).

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[3] ANSYS Inc., Innovative Turbulence Modeling: SST Model in *ANSYS CFX*, Canonsburg, PA, ANSYS, 2014.