Performance of a New Stator-Diffuser Design for an Axial-Flow Pump Unit

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Abstract

In an axial-flow pump unit with conventional stator and diffuser, often considerable energy is still present in the swirl (rotation) of the liquid leaving the stator. This energy will eventually be lost from the pump system. In this experimental investigation a new design, combining the stator and diffuser together into a single component, was tested for its effectiveness in recovering this energy and thereby improving the performance of an industry-sized single-stage axial-flow pump unit. Measurements of static pressure rise along the new stator-diffuser and of the swirl angle of the fluid leaving the pump unit indicate that the new design performs better than the conventional one, as a component. However, efficiency of the whole pump unit is in general slightly reduced with the new design. A number of factors were identified as contributing to this performance degradation. Most notable are the poor matching of the fluid's and vanes' angles at the component's inlet and the sudden expansion of the flow geometry at the component's outlet. It is thus expected that when these factors have been adequately addressed, the new design should improve the pump's overall performance.

Introduction

The purpose of this paper is to report on the performance of the new stator-diffuser designed for the axial flow pump. The new design was assessed by comparing standard performance characteristics of the conventional pump configuration with available manufacturer's curves. During this standard pump performance testing, measurements of the residual swirl at the outlet of the pump, as well as the hydraulic grade line (HGL) along the pump, were taken in order to provide in-depth information on the kinetic energy recovery process through each of the pump components.

Firstly, a general description of the test rig will be given, along with a comparison of the conventional stator and the new stator-diffuser. Discussion of the flow paths for each unit will be followed by a description of the test methodology. It will be shown that the quality of the impeller and stator blade finish has a significant influence on the overall pump performance.

The new design will then be critically assessed based on results obtained during performance testing and measurement of HGL and residual swirl. It will be shown that the new design performs better than the conventional one, as a component, even though overall pump performance is reduced. Issues with the new design will be identified and ways of resolving them will be discussed.

In general it will be shown that the new stator-diffuser meets its design objective.

Experimental Facility and Methods

The experiments were conducted on a horizontal, closed-circuit, axial flow pump test rig located in the Hydraulics Laboratory at the University of Technology, Sydney. The tested pump is an industrial size (410mm pipe diameter), single stage unit, designed by Mr Hugh Nelson [3] according to the work of O'Brien and Folsom [4] and originally manufactured in Australia by Ornel Pumps. The pump was tested in two configurations: with conventional stator and conical diffuser; and with the new stator-diffuser. Both configurations were tested with 5 and 6 blade impellers, whose blade foil sections and detailed geometry are not included here for reasons of space, but are readily available from the authors. Figure 1 shows diagrammatically the rig set up with the conventional pump design and figure 2 shows a photograph of this test rig.
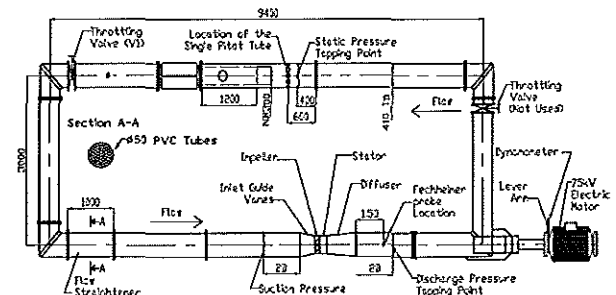

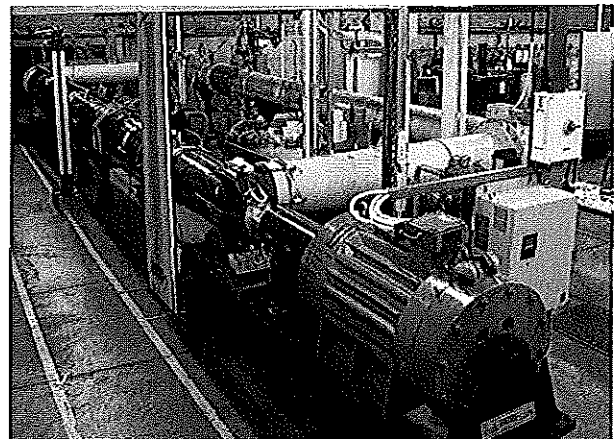Figure 1. Laboratory test rig set up diagram.

Figure 2. Photograph of the laboratory test rig.

Water is conducted from the inlet pipe with no pre-swirl to the impeller via the inlet guide vanes (note the straightener with PVC tubes in figure 1). In the conventional design, upon leaving the impeller the flow enters the 8-vane stator which is supposed to restore the flow to the axial direction. The stator is followed by a conical diffuser which reduces the velocity (kinetic energy) of the moving fluid and at the same time increases its static pressure (pressure energy) by increasing the cross-sectional area. In the new design which is investigated in this paper, the stator and diffuser are combined into a single unit as shown in figure 3. Unlike the conventional design, the new stator-diffuser has a bell shape.
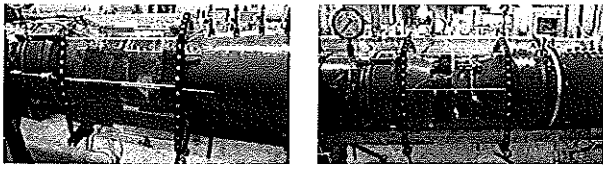
Figure 3. Pump assembly with (at left) conventional stator and conical diffuser and (at right) the new stator-diffuser.

This new stator-diffuser was designed by Mr Hugh Nelson and donated to the University by Orbit Pumps (current manufacturer of the Ornel pumps) in 1999. The design was patented in 1982 by Mr Nelson, who describes in his patent documentation the flow paths through the old and new designs of the stators.

In the conventional design of the stator, in the form of a straight cylindrical bowl with eight vanes, flow entering the stator is redirected into the axial direction by the curved blades. The trailing edge of the blades is positioned nearly parallel to the axial direction, and it guides the flow to the outlet of the stator. Mr Nelson claims that this configuration unfortunately introduces reverse flow between blades on the discharge side of the stator. This internal recirculation contributes to cavitation [2, 3] and swirling in the flow entering the diffuser, making the kinetic energy recovery process less efficient.

The new stator-diffuser has a bell shaped bowl and hub with eight vanes. The axis of the passage between adjacent blades is rectilinear and also divergent at a diffusion angle which varies between 2° and 8°. According to Shepherd [7], this is the best diffusion angle to provide efficient conversion of kinetic into pressure energy. After this conversion has taken place at the upstream section of the blades, the flow is deflected into the axial direction. Hence minimal swirling should be present in the flow leaving the new stator-diffuser.

The new design was assessed by comparing the standard performance characteristics of the conventional pump configuration with available manufacturer's curves. During standard pump performance testing, measurements were taken of the residual swirl in the flow at the outlet of the pump, as well as the hydraulic grade line (HGL) along the pump, to provide in-depth information on the kinetic energy recovery process through each of the pump components.

Flow rate was calculated from velocity measurements taken with a water manometer and a single Pitot-static probe at various radial positions across the pipe, at a location where the flow is purely in the axial direction. Selection of the radial positions was based on the requirements of ISO3966.

A mercury manometer was used to measure differential pressure across the pump (pump head) and also to take static pressure readings for HGL. To obtain HGL across the pump, differential pressures were measured between pump suction and each tapping point located along the pump assembly.

Residual swirl in the flow at the outlet of the pump was measured with a pressurised water manometer and a Fechheimer probe, manufactured according to a description given in Parmigiani et al. [6]. The probe was installed at 1.5D downstream from the diffuser's exit. Readings were taken at each performance duty point at three radial positions.

Results and Discussion

Pump performance testing was conducted in accordance with the requirements of AS2417-2001. During all tests, the pump was operated at 1465rpm, the speed for which the manufacturer's curves were originally published. Results obtained during performance testing, for five and six blade impellers with the conventional stator and conical diffuser, were in good agreement

with the manufacturer's published curves. The results obtained during the previous two studies [5, 1] conducted on this test rig with the 5 blade impeller, did not show consistency with the manufacturer's data. It is believed that the sudden improvement in results from recent testing can be ascribed to the cleaning of calcium deposits from the impeller and to the full refurbishment of the stator.

Comparison of the results obtained during performance testing with five and six blade impellers and the new stator-diffuser against the manufacturer's published curves showed that the new stator-diffuser slightly underperforms. However it is interesting to observe that during a test performed with the 6 blade impeller and the new stator-diffuser, the unit outperformed the standard stator by about 2%. This could be due to measurement error or to the fact that this was conducted on a freshly painted stator-diffuser. Figure 4 shows this test's results, where manufacturer's data are shown as continuous lines while test results as scattered points connected by line segments.
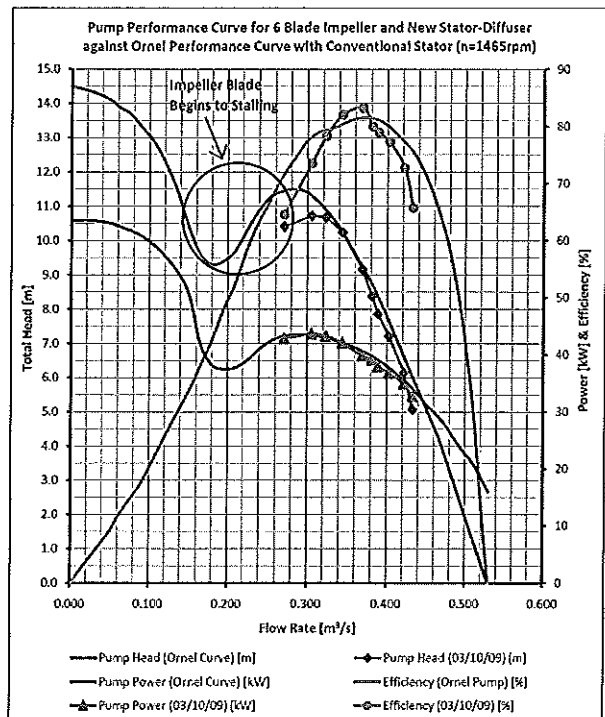

Figure 4. Pump performance curve for 6 blade impeller and stator-diffuser against manufacturer's curve.

It can also be observed from figure 4 that as flow rate is reduced beyond the best efficiency point (BEP), larger losses occur with the new unit. Pump head is about 1m lower at beginning of stall. Such losses were not detected during testing of the 6 blade impeller with the standard stator. This is believed to be due mainly to (a) the poor matching of the impeller blade exit angles with the stator blade inlet angles, causing shock losses at the stator's entrance, and (b) the sudden expansion of the flow geometry at the new unit's outlet, causing large exit loss. Another possible cause is the poor quality finish of the stator-diffuser casting resulting in high skin friction losses and a negative effect on overall pump performance. Stepanoff [8] claims that axial flow pumps are very sensitive to the quality of blade finish.

During performance testing, the hydraulic grade line (HGL) throughout the pump assembly was measured in order to evaluate static head rise across each pump component. Graphs of HGL data for the 5 blade impeller with standard stator, as well as with the new stator-diffuser, are shown in figures 5 and 6 respectively.

Results are plotted for various flow rates expressed as a percentage of the best efficiency point (BEP) flow.
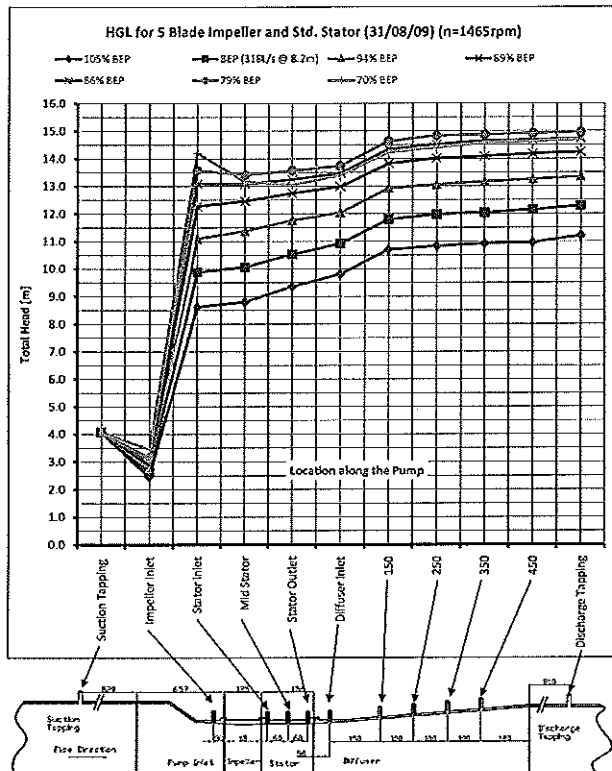

Figure 5. HGL through the pump with 5-blade impeller, standard stator and conical diffuser (n=1465rpm).

From figures 5 and 6, it can be observed that, as flow rate is reduced down to BEP, a larger portion of pressure energy is recovered by the new stator-diffuser compared with the existing design. At BEP, there is very little difference in performance between the two. Beyond BEP, the energy recovery process through the standard design is better than for the new one. All this is consistent with the discussion above. It should also be noted the standard design has greater length and most of the pressure is recovered by the diffuser and not by the stator vanes.

Also, head rise through the impeller is smaller for the new design compared with the original design. This is believed to be due to poor matching of the impeller blade exit angles with the stator inlet vane angles, and the poor quality finish of the stator vanes, causing high 'entrance losses', as discussed above. Therefore it is recommended that other sets of readings be taken after polishing the stator-diffuser vanes.

From figure 6 it can be observed that HGL for the new stator-diffuser reflects the shape of the component very well. A significant portion of kinetic energy is recovered by the front (vaned) section of the stator-diffuser, with some drop in pressure in the outlet section at exit of the stator-diffuser. Although pressure is recovered further in the discharge pipe (up to the discharge tapping point), for the 6 blade impeller (at BEP), the drop is equal to nearly 10% of the total head rise across the pump. This is caused by a step expansion in the geometry at the outlet of the stator-diffuser, which contributes to the generation of eddies as shown on figure 7. This flow pattern is verified by the results obtained from residual swirl measurements described below.
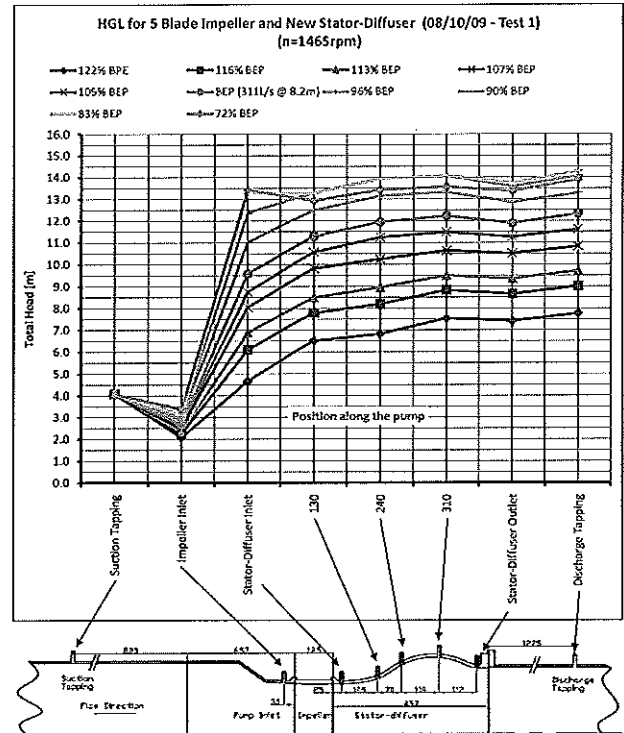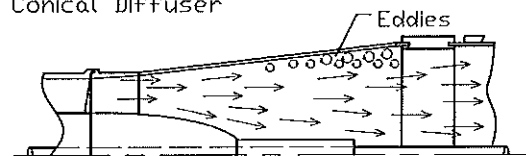

Figure 6. HGL through the pump with 5-blade impeller and new stator-diffuser (n=1465rpm).
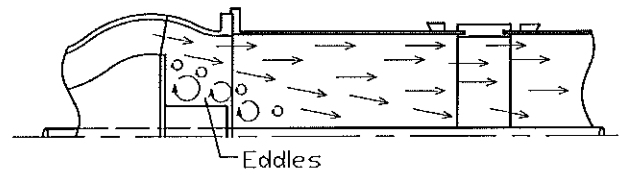

Figure 7. Theoretical flow through the diffuser and new stator-diffuser.

Figure 8 shows results of the residual swirl measurements for the 5 blade impeller with standard stator and conical diffuser, and figure 9 shows readings taken for the 5 blade impeller and the new stator-diffuser. Data at BEP are marked in black.

Comparing figures 8 and 9, it can be observed that for the conventional pump design swirl is larger at locations further from the pipe centre. On the other hand, for the new design swirling angle does not change significantly with depth (depth here is distance from the pipe wall). Although the averages can still be seen to follow the same trend as for the conventional design, it can be observed that the angles are smaller. Therefore it can be concluded that the new design is more effective in recovering the swirl, and can thus be said to meet its design objective.

Also comparing figures 8 and 9 shows that results obtained for the new design are a lot more scattered. This is due to eddies created in the step expansion in the geometry at the outlet of the stator-diffuser as shown on figure 7.
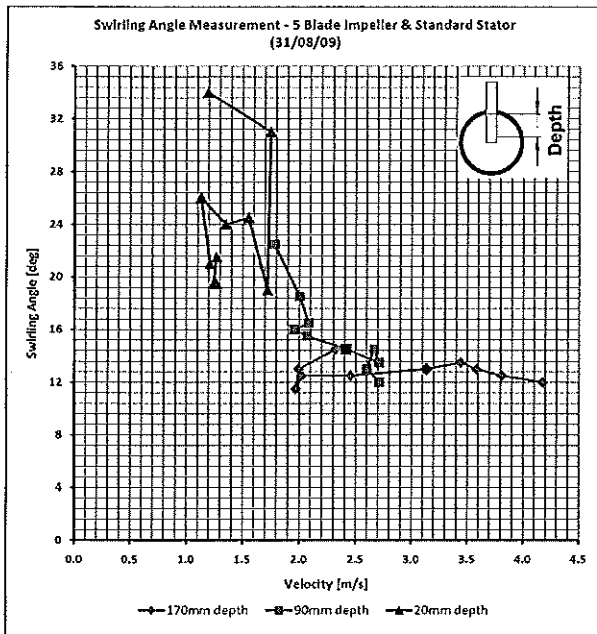
Figure 8. Fecheimer probe data for 5-blade impeller with standard stator and conical diffuser (n=1465rpm). Data at BEP are marked in black.

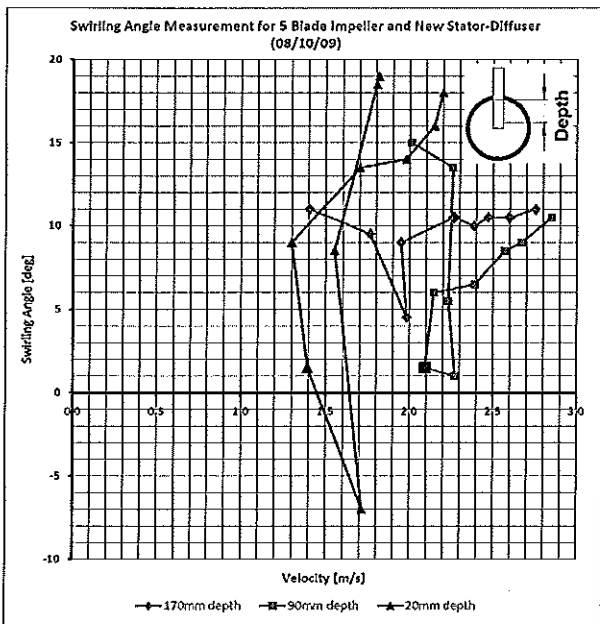

Figure 9. Fecheimer probe data for 5-blade impeller with new stator-diffuser (n=1465rpm).

Further research should be undertaken in order to improve this new design of the stator-diffuser, especially in the area of interference between impeller blades and stator inlet. It would be optimal to install a Fechheimer probe at the inlet to the stator-diffuser to check how well the angle of the fluid leaving the impeller matches the stator inlet angle. Additionally, another set of tests should be conducted on this new part after removing all casting defects.

Conclusions

Although comparison of the pump efficiency curves for the new stator-diffuser and the existing design showed that the new design is slightly less efficient (about 2%), some interesting characteristics were found during evaluation of the swirling angle

and HGL data. In comparing HGL data for the two designs, it was found that as flow rate is reduced the stator-diffuser recovers significantly more pressure energy up to the best efficiency point (BEP); however, beyond BEP, it worsens.

Furthermore, evaluation of the swirling angle data for the new stator with the 5-blade impeller confirmed that the angle systematically reduces up to BEP where it reaches a value of about 1° to 2°. Once the pump performance moves away from BEP, flow begins to swirl again. Conversely, the swirling angle in the conventional design increased proportionally throughout the entire performance curve. Based on the above it can be concluded that the new stator-diffuser meets its design objectives. However, overall performance is not optimal.

To improve the whole pump's performance, the new stator-diffuser should be modified to provide better guidance to the fluid leaving the vaned section at the outlet, as suggested on figure 10. Further investigation should also be conducted so as to improve the matching of flow angles at the unit's inlet.
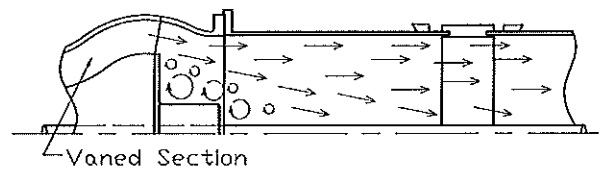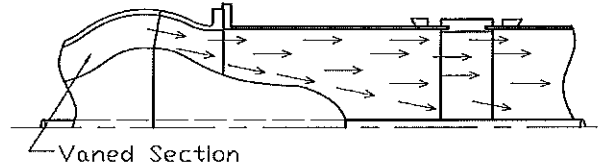
Figure 10. Proposed design modification for the new stator-diffuser.

Acknowledgments

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Welcome

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Welcome

The Organising Committee of the 17th Australasian Fluid Mechanics Conference would like to take the opportunity to welcome you to New Zealand, to Auckland and to the 17th meeting of this conference.

The Australasian Fluid Mechanics Conferences have been held triennially since the series inception as the Australasian Hydraulics and Fluid Mechanics Conference at the University of Western Australia 1962. This Conference is the 17th meeting, the 4th of the series to be held in New Zealand, and the 3rd to be held at The University of Auckland. These conferences, at which all contributions are presented orally, have the tradition of providing graduate students with an opportunity to interact with a community of richly experienced researchers, and have played a very significant role of nurturing fluid mechanics research in Australasia and the surrounding region for nearly four decades.

The role of developing this vibrant community has been formalised with the recent formation of the Australian Fluid Mechanics Society, and this is the first event which has been associated with the AFMS. A highlight of this meeting will be the celebration of the induction of the 10 inaugural AFMS Fellows at the conference banquet.

A feature of the AFMC series is the broad range of fluid mechanics research presented, and this conference is no different as evidenced by the 28 themes identified in the programme, the 235 presentations and over 200 papers submitted. The flavour of a particular meeting is set by the plenary speakers and the conference committee is very grateful for the contributions of David Boger, Vladimir Nikora, Mike O'Sullivan, Andrew Pollard and Phil Schwarz for providing high quality thematic foci.

As Chair I would like to acknowledge here the efforts of John Cater the conference secretary and the other members of the conference Organising Committee; Rosalind Archer, John Chen, Richard Clarke, Stephen Coleman, Richard Flay, Bruce Melville, Roger Nokes, Stuart Norris, Peter Richards and Rajnish Sharma. I also acknowledge the 120 reviewers of the abstracts and papers. The Committee would particularly like to thank Tessa Hagemann, of the Conference Management unit in The University of Auckland's Centre for Continuing Education who has tirelessly provided the administrative support that has allowed the committee to concentrate on the technical aspects of the conference.

Finally, the Committee would also like to acknowledge the support of the Faculty of Engineering at The University of Auckland and the Conference sponsors: the platinum sponsor Olympus, COMSOL Multiphysics, LasTEK & LaVision, LEAP Australia, Kenelec Scientific, Duff and Macintosh, New Spec & Spectra-Physics.

Gordon Mallinson
Chair

17AFMC Programme Timetable

Tuesday

09.00-10.00 Plenary: Andrew Pollard

10.00-10.30 Coffee

10.30-12.30 Technical Session 3

	CFD III - Heat Transfer, Plenary Theatre 260-098		Aerodynamics I, Case Room 2		Medical Flows, Case Room 3		Hydraulics, Case Room 4		Jets, Wakes and Boundary Layers I, Theatre OGGB3
Paper	216	Numerical Simulation of a Natural Ventilation Flow with a Line Heat Source Using Various Advection Schemes	110	Local corotation of line segments and vortex identification	114	Pressure losses in neonate breathing	148	Flow Convergence At The Tip And Edges Of A Viscous Dam Break Wave	81	Flow around a square cylinder with a detached downstream flat plate
Presenting Author		Ms Tae Hattori		Dr Vaclav Kolar		Mr Luke Sinclair		Assoc Prof. Tom Baldock		Mr Mohamed Sukri Mat Ali
Paper	152	Two- and Three-Dimensional Simulation of Combined Natural Convection Cooling of a Drink Can.	236	Experimental study of vortex meandering	280	Flow through a Model of a Non-Axisymmetric Stenosis	320	An experimental investigation of pressure loss in canvas fire hose	142	Stabilisation of the von Karman alley in a stratified cylinder wake
Presenting Author		Dr Sujin Jiracheewanun		Dr Thomas Leweke		Dr Martin Griffith		Dr Ba Phuoc Huynh		Dr Patrice Meunier
Paper	239	Effect of under-floor blockages and perforated tile openings on the performance of raised-floor data centres	329	An Investigation of Three-Dimensional Effects on the Performance of Tubercles at Low Reynolds Numbers	349	Steady inflow through a model aneurysm: Global and transient stability	347	Advanced Particle Tracking for Sediment Movement on River Beds: A Laboratory Study	188	Progress of Laminar-Turbulent Transition in a Two-Dimensional Mixing Layer (Quantitative Representation of Transition Process)
Presenting Author		Mr Babak Fakhim		Ms Kristy Hansen		Dr Gregory Sheard		Miss Katherine Heays		Mr Shinya Kamada
Paper	248	Natural convection flows in reservoir sidearms using Large Eddy Simulation and an Immersed Boundary Method	376	A new velocity scale for turbulent boundary layers with adverse pressure gradient.	193	Flow past a thrombus analog within an in-vitro environment	354	Evolution of unconfined turbidity current deposits: an experimental study	232	Flow Control Based on Three Dimensional Wake Instabilities for Reduction of Fluctuating Aerodynamic Forces of a Blunt Trailing Edge Profiled Body
Presenting Author		Mr Karl Ditko		Mr Syed Imran Shah		Mr Chris Butler		Theo Sangster		Prof Horia Hangan
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Presenting Author				Dr Stefan Schmidt		Mr Patrick Geoghegan		Mr Dean Gioslo		Prof Jonathan Paul Gostelow
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Presenting Author				Mr Chris Ellis		Mr Callum Spence		Ms Anna Czlonka		Dr Horia Hangan

12.30-13.30 Lunch

13.30-15.10 Technical Session 4

	CFD IV - Structure and Stability, Plenary Theatre 260-098		Aerodynamics II, Case Room 2		Computational / Experimental Turbulence I, Case Room 3		Oceanography, Case Room 4		Jets, Wakes and Boundary Layers II, Theatre OGGB3
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Presenting Author		Mr David Smith		Mr James Brett		Mr Bagus Nugroho		Dr Marshall Ward		Prof Min Seong Chong
Paper	209	On the Stability of Time-Periodic Pipe Flow	12	Detached-Eddy Simulation of the Dynamic Loads of C-130H with Open Cargo Bay	94	New drag balance facility for skin-friction studies in turbulent boundary layer at high Reynolds numbers	208	Shallow wake flows and the shallow-water approximation	72	Scaling of inner and outer regions for flat plate boundary layers
Presenting Author		Mr Jon Nebauer		Dr Stefan Schmidt		Mr Murali Krishna Talluru/ Mr Charitha da Silva		Dr Graham Hughes		Dr Romain Mathis
Paper	308	Stability of Steady Flow Through a Corrugated Pipe	250	Analysis of Ornithopter-Wing Aerodynamics	107	Prediction of free-stream turbulence variation around the leading edge of an axial compressor blade	258	Modelling tsunami generated by submarine landslides	100	Constant Adverse Pressure Gradient Turbulent Boundary Layers
Presenting Author		Mr Sean Loh		Miss Aliya Valyff		Mr Samuel Perkins		Dr Roger Nokes		Mr Zambri Harun
Paper	334	Early Transition of Separated and Reattaching Flows	210	Effects of Aeroelasticity on Flapping Wing Propulsion	112	Conditional large-scale structure of a high Reynolds number turbulent boundary layer	356	Wave runup investigation on a square cylinder	196	Evolution of zero pressure gradient turbulent boundary layers
Presenting Author		Prof Mark Thompson		Ms Sheila Tobing		Dr Nick Hutchins		Prof Krish Thiagarajan		Mr Vigneshwaran (Vicky) Kulandaivelu
Paper	22	Direct numerical simulation of 3D steady streaming induced by Honji Instability	247	Validation of Mean Lift Estimates for Normal Hovering Flight			34	Unsteady Turbulence in Tidal Bores: Ensemble-average or VITA?	206	Effects of Free-Stream Vorticity on the Blasius Boundary Layer
Presenting Author		Dr Hongwei An		Mr James Harvey				Prof Hubert Chanson		Mr David Pook

15.10-15.40 Tea