Efficiency of an aerator driven by fluidic oscillation.
Part II: Pilot scale trials with flexible membrane diffusers.
William B. Zimmerman¹, Vaclav Tesar², H.C. Hemaka Bandulasena¹, Olumuyiwa A. Omotowa¹
¹Department of Chemical and Process Engineering; University of Sheffield, Sheffield S10 2TN
²Institute of Thermomechanics of the Academy of Sciences of the Czech Republic v.v.i., 182 00 Prague, Czech Republic.

Abstract
Pilot plant aeration performance with microbubble generation by a novel fluidic oscillator driven approach is studied, with a view to identifying the key design elements and their differences from standard approaches to wastewater treatment. The microbubble generation mechanism has been shown to achieve high mass transfer rates by the decrease of the bubble diameter, by hydrodynamic stabilization that avoids coalescence increasing the bubble diameter, and by longer residence times offsetting slower convection. The fluidic oscillator approach also decreases the friction losses in pipe networks and in nozzles / diffusers due to boundary layer disruption, so there is actually an energetic consumption savings in using this approach over steady flow. These dual advantages make the microbubble generation approach a promising component of a diffuser network for wastewater treatment. The clear water trials show that there is a best configuration for fluidic oscillation to accelerate the oxygen transfer efficiency in the initial stages of transfer, particularly while the DO concentration level is less than 3mg/l which is the industrially important regime. One measure of the accelerated oxygen transfer, the dosage, shows a 3-4 fold improvement over the control and the standard design, respectively. The best power consumption decrease found in this oscillatory configuration is 18%. Furthermore, this configuration is at 83% of the designed volumetric air flow rate for conventional operation of the diffuser network.

§1 Introduction
In Part 1, we introduced a new technique for generating microbubbles from apertures oriented appropriately in a solid nozzle bank so that the detached bubble is approximately the size of the diameter of the aperture by exploiting fluidic oscillation. Several designs were trialled visually, and the most promising for operational purposes was found to be the dynamic mode of high frequency oscillation where the bubble detaches due to the overpowering by the inertial acceleration of the pulse of air against the “anchor force” of wetting of the aperture perimeter. This mode was shown to produce a nearly monodisperse, uniformly spaced, and non-coalescent cloud of bubbles. The bubble cloud so produced was at least seven-fold better at oxygen transfer from a combination of longer rise time and the product of higher mass transfer coefficient and surface area per unit volume of the gaseous phase than the control of steady air flow. Hwang and Stenstrom (1985) claim that 60-80% of the energy costs from wastewater treatment are expended on aeration, so the potential for cost savings is compelling.

An open question, however, is whether the oscillatory flow has an attendant higher friction loss, thereby potentially offsetting the performance gains. Furthermore, common industrial practice in wastewater aeration is to use bubbles emerging from diffusers – typically disc membranes with a flexible membrane patterned with millimetre scale slits that open and close to create small bubbles. Whether the membrane slits can be used in coordination with fluidic oscillation has not been previously tested.

Wastewater aeration can be accomplished by a wide range of technologies (see Tchobanoglous and Burton, 1992), including surface aerators (Rao and Kumar, 2007) which rely on turbulent entrainment from rotating blades. However, flexible membrane slit diffusers fed by blowers are considered the modern and efficient approach in wastewater treatment for aeration by bubble dispersal. If there is a benefit from using a fluidic oscillator
system in conjunction with flexible membrane diffusers, then it can be incorporated by a change of the pipework to include the fluidic circuitry above water after the blowers.

In this paper, we test the hypothesis that fluidic oscillation can lead to a benefit in wastewater aeration by enhancing mass transfer performance and more efficient oxygen transfer, accompanying the likely generation of smaller bubbles. In order to test this hypothesis, in §2 we describe the design of a fluidic oscillator with response in the control flow designed to match the load of the two nine inch standard diffusers connected to the outlets. The steady, turbulent computational fluid dynamics simulation of two different oscillator geometries that are topologically equivalent, but achieve different pressure difference from the terminals and the air flow rate through the control loop. In §3 we report on the outfitting of a pilot plant with a pipework array feeding two banks of eight membrane diffusers each, fed by the outlets of a fluidic oscillator. The pilot plant is instrumented with a volumetric flow meter upstream of the fluidic oscillator and a control valve to regulate the flow, downstream of the blowers. The blowers are instrumented with an ammeter to assess power draw. In §3, a range of volumetric flow rates and oscillation frequencies are used to operate the plant, and compared with the control of steady flow through a tee splitter into the diffuser pipework banks. The power draw is assessed. An underwater camera is used to visualize the bubble dispersal dynamics. The audio track is analyzed to estimate the oscillation frequency achieved with the given load of hydraulic resistance on the fluidic circuit – pipework, diffusers and head of water in the tank.

In §4, a selection of the flow rates and frequencies sampled in §3 are used in conducting aeration trials from deoxygenated clean water while dissolved oxygen levels in the tank were recorded. These profiles are analyzed. In §5, discussion and conclusions are presented.

§2 Design of a fluidic oscillator for flexible membrane diffuser bubble generation

The microbubble generator described in Part 1 used a simple aerator nozzle bank, exploiting the dynamic, high frequency mode that limited the growth of the bubbles to approximately the same size as those of the apertures in the nozzle bank. The key factor for the success of this mechanism is a reliable bubble detachment at the end of each oscillation period – not allowing them to retreat back into their apertures.

A suitable detachment mechanism may be provided by using standard membrane diffusers. Figure 1 shows a schematic of a detail of a part of the membrane. It is provided with cut slits, which open when the membrane is deformed by the air pressure as in the right frame of Figure 1.

Figure 1    Schematic drawing of a part of the membrane with slit cuts. At left basic state – no pressure difference. At right: membrane bulged by the acting air pressure. The parts "z" between the slits are not tensioned in the vertical direction and this is why the cuts open.
Figure 2. Schematic a bubble emerging from the membrane during an applied pressure pulse. In b the pressure exceeds membrane resistance as well as the hydrostatic pressure and the slit opens, c. In d the bubble is necked not only due to pressure pulse ending but also because the surface tension on the bubble surface decreases. In e, the membrane retracts and detaches the bubble.

The idea behind the use of the membrane in standard steady- flow supplied aerators is the pressure difference across the membrane decreases once some air escapes to form the bubble – Figure 2 - and this decrease suffices to close the slit and detach the bubble. This, however, is dependent on precise interplay of pressure changes and in our experiments was not found to be an absolutely reliable mechanism – the pressure decrease due to bubble escape may be not sufficient so that the slit does not close and the bubble continues to grow. The operation is also dependent on the shape of the membrane (i.e. on the pressure difference level) prior to the opening of the slit.

The working hypothesis of the present authors is the operation – opening and closing – of the slits in the membrane may benefit from the air supply pulsation provided by the fluidic oscillator – and, on the other hand, the membrane may provide the detachment mechanism missing when the oscillator is used with simple exit orifices. The aggregate layout brings less dependence on the precise pressure levels adjustment, which is usually not easy to maintain in the difficult conditions in aeration plants.

The fluidic oscillation can actually interfere with the dynamics of the membrane slit bubble formation in several ways. The membrane is always under pressure due to the steady flow component leaving the oscillator. One possibility is to apply very slow oscillation together with the jet-pumping effect in the OFF output terminal. The membrane is thus periodically relieved from the extension all the way back to the flaccid, undeformed state as shown in Figure 1 on the left frame (or in Figure 2 on the frame a). The oscillation then sets up a mechanical motion of the membrane with repeated “puffing out” and “deflating” phases that also deform the slits in a regular uniform phase across the entire membrane, rather than randomly as one slit opens and closes. This uniformity enforced across the membrane motion should create regularly spaced uniformly ejected bubbles, which should be hydrodynamically stabilized by the mechanism described by Crabtree and Bridgwater (1969) and therefore non-coalescent.

Alternatively, we can consider the possibility of very rapid oscillation, which should not give the membrane much time to “deflate” from its nearly fully extended position. Rapid oscillations should be viewed as a series of high momentum pulses with an acceleration inertial force proportional to the frequency. Thus the pulses are inherently strong, and the acceleration due to their transient nature can provide an additional momentum. Conceivably, the arrival of a strong pulse of air would force open the slit more rapidly than the standard steady flow mechanism, but then, the finite capacity of pulse-flow momentum ceases to hold the slit open. The time scale for opening and closing is not, then, the “natural” time scale of
the steady flow operation which is determined by surface tension induced channelling, similar to the famous Saffman-Taylor viscous fingering (see Zimmerman and Homsy, 1991).

Another important feature is the frequency of the opening and closing of the membrane slits themselves. With the steady air flow mechanism, it is determined by the bubble interfacial dynamics. However, it is well known that flexible systems behave like classical spring oscillators – given the slit size and the Hooke’s law spring constant (modulus) of the material, there is a natural frequency of oscillation of the slit. Due to the viscosity of the fluid and the friction losses of the material, one would expect that this natural mode is damped and of little concern. However, the fluidic oscillation makes this a forced, damped nonlinear oscillator. Such systems are well known to show resonant effects when forced with the natural frequency of the oscillator (see Zimmerman, 2006) or a harmonic of this fundamental frequency of excitation. In these cases, it is expected that the slit membrane would open and close at the resonant frequency. This should be matched by the frequency the oscillator supplying the air pulses to the membrane. Again, in this case the time scale for bubble generation would be selected by the natural mode of the slit membrane, not the surface tension driven process. This period could be faster, in which case smaller bubbles are produced.

To test these hypotheses for the beneficial use of fluidic oscillation, we have designed, made, and tested a fluidic oscillator of unusually large scale - matched to the expected flow rate regime in the inlet and to the loads placed on the outlets. In principle, the oscillator followed the earlier small-scale designs, as described in references (Tesar et al. 2006,2007; Tesar 2009a,b,c), with which was accumulated positive experience. Also the manufacturing technique of making the devices – cutting the cavities by laser in PMMA plates – was retained.
The oscillator was again designed to consist of a fluidic amplifier provided with feedback loop. The amplifier is of the jet-deflection type, bistable — with bistability due to the

Figure 4  Detailed photograph of the core part of the fluidic amplifier. The cavities – later cover by flat top and bottom cover plates – were laser-cut in a stack $5 \text{ mm}$ thick of PMMA plates.

Figure 6  Photograph of the assembled amplifier with the 1 inch supply and output terminals provided with quick-release hose connectors.

Coanda-effect attachment to one of two attachment walls located opposite one another on both sides of the space through which passes the main jet issuing from the supply nozzle. Its planar geometry, is shown in Figure 3. The next Figure 4 is a detail view of the key part, with the supply nozzle flanked by the two control nozzles. The cavities were made in $5 \text{ mm}$ thick PMMA plates, stacked on top of one another. They were closed from top and bottom by flat cover plates, of $15 \text{ mm}$ thick PVC – seen clearly in Figure 6.

Essentially, the geometry is a scaled-up version of the amplifiers described in Tesar et al. (2006, 2007). The main difference is the considerable curvature of the diffusers leading to the output terminals. This is, in general, not advisable since curved diffusers are prone to
flow separation from the wall on the inner side of the curvature. Here, however, the curvature radii are very large and no separation was detected. On the other hand, this layout of the diffusers allowed having them long and yet suitable from the manufacturing point of view — allowing for making two amplifiers from a single square-shaped PMMA plate of 0.4 m x 0.4 m size, as shown in Figure 7.

Figure 7 The reason for the unusual bent collector diffusers is this design’s suitability for being laser-cut from a square-shaped PMMA plate with minimal material loss.

The amplification effect – deflecting the main jet by the relatively weak outflows from one or other control nozzles – is stronger if the main nozzle exit is of smaller cross section. Also, the deflected jet remains in the desirable inclined shape, despite the hydraulic resistance which it meets in the output, if it is more accelerated in the supply nozzle. On the other hand, forcing the air flow through a relatively small cross-sectional area would lead to high hydraulic loss. It is the task of the diffuser to counter this loss by pressure recovery – re-conversion from the kinetic energy of the air. To be effective, the diffusers have to be of small divergence angle and achieving the required area ratio makes them very long.

Instead of the more obvious version of the feedback – two loops, each connecting the output terminal with the control terminal on each side of the amplifier, present authors use the less known Spyropoulos feedback, formed by a single tube or hose connecting the two control terminals (A and B in Figures 8 and 9). The measured frequency of generated oscillation – with the two connected 9inch diameter aeration diffusers with the standard membranes, as shown in Figure 8 –
The two nine inch diffusers with flexible, slit-cuts membranes, are attached – via 1 inch hoses - to the output ports of the fluidic oscillator. A and B are the terminals connected by the feedback loop.

Another view of the assembled fluidic amplifier, showing the terminals A and B to which is connected the hose acting as the feedback loop. The original stack of recognisably three PMMA plates was later reduced to only two plates with noticeable improvement in performance.
Figure 10  Measured oscillation frequency with the $l = 1.45$ m long feedback loop hose as a function of supplied air flow rate (with attached membrane diffusers).

is presented in Figure 10 as a function of the air flow rate. Results of the same experiment are re-plotted using dimensionless parameters – Reynolds number $Re$ and Strouhal number $Sh$, both computed using the main nozzle width as the characteristic length - in the next Figure 11. At higher flow rates the Strouhal number is practically constant and this invariance is an important factor for computing the performance parameters in various alternative conditions – especially larger air flow rates that may be required - arising in various aeration applications.

Figure 11  Dimensionless presentation of the oscillator frequency by means of Strouhal number $Sh_b$ evaluated using the main nozzle width $b$ as the characteristic dimension.
Of course, the value of the Strouhal number depends on the length of the feedback loop hose - in experiments with essentially geometrically similar case in Tesar et al. (2006), the oscillation frequency was found to be simply inversely proportional to the feedback loop length $l$.

Originally, the task was to drive by the oscillator just two aeration diffusers – in a manner shown in Figure 8. The dimension of the diffuser inlet – a 1 inch diameter threaded connector – thus dictated the size of the oscillator, having also the supply as well as output terminals of the 1 inch diameter size. While suitable for the initial tests, this size of the oscillator was later considered too small for use in practical aeration applications. Also, a larger size leads to lower pressure drops, roughly in inverse proportion to the fourth power of ratio of diameters. In spite of most of the pressure difference to be overcome by the air source being the relatively high aerodynamic resistance of the membrane aerators – as well as the hydrostatic pressure – due to the aerator exits locate at the bottom of the aeration tank – decreasing the required air pressure is considered an important factor for improving the economy of water treatment. As a result, there was a demand of scaling the whole existing design up, in ratio 2:1, i.e. for 2 inch (roughly 50 mm) diameter of the supply and output terminals.

For this scaled-up size, it seems quite probable that the manufacturing technique of the fluidic devices actually used in the aeration plants would be different – nevertheless, for the immediate needs of tests it was decided to make even these large devices by the same method of laser cutting. Nevertheless a simple scaling would lead to too large and too

![Figure 12](image_url)  
**Figure 12**  Geometry of the later, larger planar fluidic amplifier. The size of the cavities is twice the size of the amplifier in Fig. 3 and supply and output terminals are here of the 2 inch (= 50.8 mm) diameter.
Figure 13. Photograph of the stack of thick PMMA plates made for the very large fluidic amplifier.

Figure 14. A detail of the output terminal of the very large fluidic amplifier: the scale documents the output terminals are here of the 2 inch (= 50.8 mm) diameter.
expensive objects and it was decided to adjust the geometry of the large devices – as it is shown in Figure 12. It is immediately apparent that the diffusers are now shorter.

Details of the very large plates used in a stack as the key parts of the very large amplifier (covered again an top as well as bottom by flat cover plates with connecting terminals) are presented in Figures 13 and 14. In principle, this very large amplifier may be provided with feedback loop hose scaled up in the same ratio as the rest of the device. However, especially in one of the above mentioned operation modes with very low frequencies, this hose would be inconveniently long. It was found preferable to use the already existing oscillator as described above and use it as the pilot device, fully controlling the flow switching between the aerator diffusers as shown in Figure 15.

§3 Aeration efficiency study: layout and energetics

Deep water diffuser aeration systems have long been hailed as a way of increasing the oxygen transfer efficiency by improving the residence time of the conventional, larger bubbles produced by diffusers. By increasing the head of water above the diffusers, there is a longer residence time for the bubbles to transfer their oxygen content. It is a clear inefficiency in conventional diffuser aeration systems that bubbles violently burst at the top surface, releasing their oxygen content with still a large untransferred amount. Deep water diffusers have the potential to consume more of the contents. Smaller bubbles have this same potential in conventional diffuser aerators, as the bubbles have a shorter rise time in general, with the rise velocity approaching the Stokes terminal velocity from above as the bubbles exit with greater momentum than just the buoyancy driving force. Deep water diffuser systems, however, suffer from the difficulty of finding appropriate operating conditions to overcome the drawback of higher pressure head on the bubble formation point opposing the bubble formation and requiring higher blower pressure at a minimum.
The problem of translating the laboratory results of Part 1 and §2 to the pilot scale is two-fold, and very similar to the problem of developing a deep water diffuser system: (1) the pressure head is higher, and may influence the interaction of the three forces that determine the bubble detachment mode in Part 1 – air pulse momentum/acceleration force, wetting anchor force, and buoyant force. Potentially, it is only a question of having sufficient air pulse momentum or acceleration force to overcome the combined additional pressure head and the wetting force, but it may be more complicated. Finding the dynamic mode of operation is an experimental question since the theory of bubble detachment dynamics and associated modelling and simulation are not sufficiently advanced yet to address the question; (2) the detachment dynamics are different for the fixed nozzle banks of Part 1 than the flexible membrane diffusers discussed in §2.

In this section, we address the bubble dynamics and energetics in two ways. The pilot plant experiments are described, which permits the characterization of the power drawn in steady operation. Underwater visualization studies are used as a guide to the mode of operation, and are really only qualitatively described. The facility used was outdoors and the site was not particularly secure. For instance, thieves stole the main blower and the spare was used for the trials. We therefore did not wish to “permanently” mount the underwater video camera for fear that it would join the main blower. Also, due to initial lack of knowledge about the flow regimes and the distribution in the system, we thought that the visualization studies would be most informative if the camera were to “roam.” However, this decision makes quantitative analysis of the images, such as attempts at bubble size distribution, too inaccurate to be useful.

§3.1 Pilot plant description
A purpose-built pilot plant unit was used for trials of the fluidic oscillator driven microbubble generator through standard 9 inch diffusers with a flexible membrane supplied by Suprafilt. The major element of the pilot plant unit is a tank with internal dimensions (2m by 4m by 3m deep). This tank was laid out with two banks of 8 diffusers each (see Figure 16), fed air through the two outlet terminals of a fluidic oscillator system (see Figure 9,17). The inlet of the oscillator is delivered air from a blower system with a volumetric flow meter installed. The blower electric current usage can be measured by drawn amperage with a constant AC voltage imposed. The tank was instrumented with a single dissolved oxygen probe at the geometric center of the tank, relative to the rectangular base, and at the mid-height 1m above the tank floor in 2m of water. The dissolved oxygen probe, the YSI 556 Multiparameter System model, was manufactured by YSI Environmental, with a precision of 0.001 mg/l and the ability to measure relative to the solubility level at the water temperature.

§3.3.1 Diffuser layout
The number of diffusers and their layout in Figure 16 was designed with the standard methodology for a tank of those dimensions at the nominal flow rate of 30m$^3$/hr with the advice of Graeme Fielden of Suprafilt (personal communication).
Figure 16. Two separate banks of eight 9inch disk diffusers (Supraflit Ltd.) laid out at the bottom of 3m tank.

Table 1: Power Consumption for experiments at different flowrates and feedback loop lengths giving the corresponding current drawn.

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§3.3.2 Power consumption and flow rate

The flow rate is controlled by a ball valve which permits the turndown of flow from the \(\sim 30\text{m}^3/\text{hr}\) maximum. The blowers stall at approximately \(\sim 15\text{m}^3/\text{hr}\) unless the flow is bled off before the control valve. Table 1 shows the measured amperage drawn at different lengths of feedback loops at flow rates from 15 to 30 \(\text{m}^3/\text{hr}\) and at the control. The control experiment replaces the fluidic oscillator complex of Figure 8 with a tee splitter which feeds both banks of diffusers with a steady flow. The surprise is the decrease registered in power consumption, with highest percentage value of 18\% for the \(L=0.4\text{m}\) configuration over the control at 25 \(\text{m}^3/\text{h}\).

§3.4 Energy efficiency

One of the unexpected outcomes of an ongoing set of pilot scale trials in wastewater treatment with a pneumatic distributor system for two banks of conventional aerators, called membrane diffusers, was the decrease in power consumption by the fluidic oscillator inserted as the splitter between the two banks (see Figure 4). Typically, one expects that the insertion of a fitting into a flow distribution system, such as a bend, valve, or splitter will add an additional hydraulic resistance to the system. So the design trade-off for microbubble generation driven by a fluidic oscillator would be expected to be increased mass transfer performance scaling with the ratio of the diameter typical bubble generated by free stream steady flow and the microbubble diameter generated by fluidic oscillation, and the expected additional head loss at a constant volumetric flow rate. Clearly, the unexpected hydraulic resistance decrease of oscillatory flow requires an explanation. We believe there are three potential components to the decrease, as discussed under the next three subheadings.
§3.4.1 Coanda effect friction reduction
The fluidic oscillator in Figure 4, on time average, serves as a splitter. The aggregate flow rate through each exit channel is equal, yet the flow never flows down both channels simultaneously. If the feedback loop were omitted, i.e. the control ports closed off (Figure 4), then it would be expected that the fluid would fill the ducts on both sides due to the splitter action. This is exactly what happens in our control experiment, where the fluidic oscillator is replaced by tee-splitter. The tee-splitter and the closed-off control ports on the oscillator achieved almost exactly the same energy consumption at constant volumetric flow rate. In both cases, the splitter action results in the mean flow having a stagnation point at the geometric point of the split, regardless of the design of the splitter. Conversely, when the jet flows through either channel in the fluidic oscillator driven flow, the jet attaches to the curved sidewall in Figure 4 according to the Coanda effect. Although this flow smoothly curves toward the outlet port, the diverted jet has no stagnation point. The friction loss along the wall at and near a stagnation point is appreciable, and is completely avoided by either diverted jet in a fluidic oscillator.

§3.4.2 Bubble release dynamics
Because the forces causing bubble release are different when the pulse of air arrives with successively higher acceleration force with increasing oscillation frequency, how the bubble detaches is intrinsically different than steady flow detachment. With steady flow, the detachment process as illustrated in Figure 2 is a gradual process where friction losses would be expected to be fully realized – not just the wetting force fully dissipating contact energy of the liquid to the solid, but the transfer of momentum from air to liquid is balanced. If the air pulse, however, arrives with significantly higher momentum so that it detaches rapidly, the contact time for friction dissipation will be much shorter. The analogy is that of the ballistic motion of a bullet in two cases – the bullet that passes through a wall is slowed by the wall, certainly, but not nearly as much as the bullet that becomes embedded in the wall, which dissipates all its energy.

§3.4.3 Boundary layer effects
Turbulent flow in ducts experiences a viscous sublayer near the wall in which dissipation is largest, exceeding the interior dissipation from eddy motions in the bulk. It is well known that solid bounding surfaces induce most of the dissipation loss in statistically stationary turbulent flows. But what about oscillatory flows with the same average volumetric flow rate? Since our oscillation is a “positive displacement” synthetic jet, it is conceptually useful to view it as a series of momentum pulses separated by momentum “gaps”. The fluid is suddenly accelerated by the momentum pulse, and then its inertia trails off until the next momentum pulse is excited. A conceptual model for this is the classical boundary layer problem of the suddenly accelerated plate, for which the frame of reference is changed to the stationary plate with the fluid suddenly accelerated. The laminar result is presented in the classical work by Rosenhead (1963). The thickness of the laminar boundary layer $\delta$ and the skin friction coefficient $C_f$ are given, in dimensionless form, by

$$\frac{\delta}{x} = \frac{5}{\sqrt{Re_x}}, \quad C_f = \frac{0.664}{\sqrt{Re_x}}$$  \hspace{1cm} (1)

where $x$ is the downstream coordinate from the start of the pulse. With a laminar boundary layer at high Reynolds number, one could argue that the time to set up the boundary layer should be inversely related to the dimensionless boundary layer thickness, and thus scaling with the square-root of the Reynolds number. So the time to set up a boundary layer is large.
What if the period of the fluidic oscillator switches before the boundary layer is set up? That problem for dual laminar impingent jets has been studied by Hewakandamby (2008). The heat transfer coefficient for the oscillating impingent jet was found to be much higher than under steady dual impingent jets, as the oscillation disrupts the formation of the boundary layer that limits the transfer to the surface to conduction through the boundary layer. This principle works as well for mass and momentum transfer. The transfer rate to the impinging surface is much higher due to the disruption in setting up the boundary layer in the direction opposite of the impinging jets. Clearly, the argument works as well for a single impinging jet. Tesar et al. (2007) show a similar conclusion experimentally for turbulent heat transfer.

The speculation here is that oscillatory flow reduces skin friction since the viscous boundary layer is disrupted in forming in the direction perpendicular to the flow. The momentum pulses find much less resistance in pushing down the center of the channel than from the slower moving fluid near the wall, as the viscous friction has not had time to “diffuse” the sink of momentum at the wall.

This argument for skin friction reduction works as well for turbulent flow, but the time scales for turbulent wall boundary layer establishment are shorter in scaling factor, but given the much higher Reynolds number achievable in turbulent flow, this feature can be overcome with higher flowrates (or faster oscillation). The classical estimates for turbulent boundary thickness and skin friction are:

$$\frac{\delta}{x} = \frac{0.385}{Re^{1/2}} \quad C_f = \frac{0.0594}{Re^{1/2}}$$

Without any detailed experimental study, the results from our pilot trials suggest that these two resistance reduction effects – Coanda effect removing the stagnation point of the splitter and skin friction reduction by slow boundary layer formation – are estimated to be about equal in importance, about 6-7% reduction each with one volumetric flow rate, with the
inference based on a roughly linear decrease in energy consumption with increasing oscillation frequency at high frequencies, but a plateau in reduction at low frequencies, but too little data for a more accurate assessment. Higher flow rates led to greater energy consumption savings, consistent with the implication of equation (2) and our assumption about the scaling of the time to set up a turbulent boundary layer.

**Does pulsatile flow have inherently lower bulk dissipation?**

A flow is pulsatile if it has a regularly changing velocity driven by a regular changing pressure that is time periodic and abrupt, on and then off, with sharp variation. The most familiar pulsatile flow is the human blood circulatory system. The beating of the heart creates rapid, periodic variation of blood pressure, and thus a sharp variation between systolic (high pressure pulse) and diastolic (low pressure relaxation) flow rates. An example is shown in Figure 17.

A fluidic oscillator is a flow circuit element which can actuate pulsatile flow in a pipeline or pipe work network. Pulsatile flow can be idealized by a square wave profile as in Figure 10:

The pressure $p$ profile in Figure 10 can be succinctly expressed as

$$p = p_0 + P \left( \cos \left( 2\pi \left( \frac{ft}{4} \right) \right) > 0 \right)$$

(3)

Where $p_0$ is the ambient pressure and $P$ is the amplitude of the square wave. $f$ is the frequency of the oscillation. $>$ is a logical operator which evaluates to 1 when true and 0 when false.

The velocity, $u(z,t)$ in Figure 10, is taken to have only the axial component and depends only on the radial position. The unsteady, $z$-momentum equation in this case (unidirectional flow) is

$$\frac{\partial u}{\partial t} - \frac{1}{Re} \nabla^2 u = -\frac{\Pi}{\alpha} \nabla p$$

(4)

where we have been able to neglect the nonlinear inertia terms because they are identically zero in unidirectional flows. The flow quantities in (4) have been scaled so that $U$ and $P$ are dimensionless. The scales were selected so that there are two dimensionless parameters.

$$Re = \frac{\rho D^2 f}{\mu}; \quad \Pi = \frac{P}{\rho D^2 f^2}$$

(5)
The parameter \( \text{Re} \) is like Reynolds number but has characteristic velocity \( v = \frac{Df}{P} \). \( \Pi \) has the interpretation of a ratio of the pressure amplitude to the inertial pressure introduced by pulsatile flow, i.e. \( \rho v^2 \). Additionally, the flow must satisfy mass continuity or conservation, which mathematically is expressed as below for incompressible fluids with no density changes,

\[
\nabla \cdot u = 0 \quad \text{(5)}
\]

given that \( u = u(r,t) \), (5) is identically satisfied.

The boundary conditions are complete if the following are adopted

No slip at wall: \[ u\big|_{r = \pm \frac{1}{2}} = 0 \]

Uniform pressure at inlet: \[ p\big|_{z=0} = \Box(t); \quad p\big|_{z=\alpha} = p_0 = 0 \quad \text{(6)} \]

where \( \alpha = \frac{L}{D} \).

The reference pressure in Figure 18, \( p_0 = 0 \), is taken for convenience.

By incorporating the pressure amplitude in \( \Pi \), the pressure inlet boundary condition is just the square wave pure function with unit amplitude and period in Figure 9, which we will call \( \Box(t) \), in terms of its Fourier sine series

\[
\Box(t) = \frac{1}{2} + \frac{2}{\pi} \sum_{n=1,3,5} \frac{1}{n} \sin(n\pi t) \quad \text{(7)}
\]

Because of the assumption of \( u = u(r,t) \), it is possible to simplify the axial momentum equation by expressing the \( r \)-component dependence only:

\[
\frac{\partial u}{\partial t} - \frac{1}{\text{Re}} \frac{1}{r} \frac{\partial}{\partial r} \left( r \frac{\partial u}{\partial r} \right) = -\frac{\Pi}{\alpha} \frac{\partial p}{\partial z} \quad \text{(8)}
\]

using the fact that \( \nabla \cdot u = 0 \) with either (3) or (8) to show, assuming \( p = p(z,t) \) only, that

\[
\nabla^2 p = \frac{\partial^2 p}{\partial z^2} = 0 \quad \text{(9)}
\]

Equation (9) is Laplace’s equation, which since it does not depends on \( r \), can be integrated twice and the unknown integration constants can be fixed by the boundary conditions (6) shows the resulting pressure profile is given by

\[
p = \frac{\Box(t)}{\alpha} (\alpha - z) \quad \text{(10)}
\]

Substituting (8) into (6) gives

\[
\frac{\partial u}{\partial t} - \frac{1}{\text{Re}} \frac{1}{r} \frac{\partial}{\partial r} \left( r \frac{\partial u}{\partial r} \right) = \Pi \Box(t) \quad \text{(11)}
\]
Because the nonlinearity in the axial momentum equation (3) identically disappears, (3), (8), and (11) are all linear equations. In general nonlinear equations rarely have closed form, analytic solutions, but there is a good chance that there is a closed form analytic solution for a linear equation. One important reason is that linear equations can have solutions split into parts that do not interact, termed the linear super principle. In the case here, it is useful to decompose the solution $u(r,t)$ into a infinite number of parts

$$u(r,t) = u_0(r) + \sum_{n=1,3,5} u_n(t)$$  \hspace{1cm} (12)

With the decomposition (12), (11) splits into two types of equations:

steady:

$$-\frac{1}{\text{Re}} \frac{1}{r} \frac{\partial}{\partial r} (r \frac{\partial u}{\partial r}) = \frac{\Pi}{2}$$  \hspace{1cm} (13a)

unsteady:

$$\frac{\partial u_n}{\partial t} = \frac{2\Pi}{\pi} \frac{1}{n} \sin(n\pi t)$$  \hspace{1cm} (13b)

The steady equation is identical to the classic Hagen Pouiseuille pipe flow problem. By integrating twice and applying the no slip BC that a parabolic velocity profile results.

$$u_0(r) = \frac{\Pi \text{Re} \frac{1}{8}}{(1 - r^2)}$$  \hspace{1cm} (14)

Therefore, since at time $t=0$,

$$u_n(t = 0) = 0$$

Integration of equation (13b) yields

$$u_n(t) = \frac{2\Pi}{(\pi n)^2} \cos(n\pi t)$$  \hspace{1cm} (15)

An important question is whether pulsatile flow results in any extra friction losses. Since we have fixed the amount of energy available, we can see that steady response is identical to the steady pressure $\frac{\Pi}{2}$ applied to the interface which is the average pressure applied with the square wave forcing with amplitude $\Pi$. So any additional flow on average is a gain due to pulsatile forcing if positive, or a loss, if negative.

To evaluate this, it is necessary to take the time average of the flow,

$$\langle u_n \rangle = \frac{1}{T} \int_0^T u_n dt$$  \hspace{1cm} (16)

where $T$ is the interval over which the average is taken. The easiest integral to compute is where $T$ is an integer, since that averages over a periodic time scale, and $T=1$ is the easiest.
Figure 19: Oscillation frequency of the fluidic oscillator against volumetric flow rate as measured by acoustic signal analysis (peak-to-peak-to-peak) for four different lengths of feedback loops (L=3m,6m,15m,25m) at six different flow rates (Q=15,18,21,24,27,30 m³/hr).

This selection makes clear that the \( \langle u_n \rangle = 0 \), i.e. there is no additional pressure drop introduced by purely pulsatile flow.

§4 Aeration studies
In this section, we report on the experiments intending to test the hypothesis outlined in §2, that fluidic oscillation can improve the performance of a pipework array of diffusers, as arranged in Figure 16. In the first instance, reported in §4.1, the underwater video camera was used to explore the range of dynamic responses with the oscillator-amplifier shown in Figure 8 using various feedback loop lengths to alter the frequency and with different inlet volumetric flow rates.

One of the benefits of the video capturing of this parametric range is the ability to analyze the frequency of oscillation. When operating, the fluidic oscillator-amplifier system produces a very low frequency hum, which is barely audible, and is probably a harmonic that is heard. The frequency is more felt, like a very deep bass singer, than heard. The peak-to-peak duration can be estimated by data analysis of the audio track, and this is assumed to be the half-period of the oscillation, since it is expected that the trough of the upstroke is the peak of the downstroke, which registers an audio maximum. Therefore a complete cycle takes two peak-to-peak intervals to register a full period on the audio track. Figure 19 demonstrates the estimated frequencies from the analysis of a 60 second duration segment for each parametric value. Not surprisingly, the combination of higher flowrate and shorter feedback loop length results in higher frequency oscillation. The correlation of the lab bench scale version of the fluidic oscillator is demonstrated in Tesar et al. (2006). This graphic is essential to the scale up of the process from pilot scale to field or industrial scale wastewater aeration, as the key design information is the volumetric flow rate per diffuser (in this case divided by 16 diffusers) and the oscillation frequency that achieved the maximum oxygen transfer efficiency. This information is necessary for the design of the oscillator systems that will deliver the same conditions at the diffuser on the larger scale pipework-diffuser network.

Section §4.2 reports on the aeration performance of the oscillator system with parametric variation in comparison with the control of steady flow through a tee junction.
Figure 20. Underwater visualization sample. A large dynamic range of flowrates and feedback loop configurations used for the characterization of the power drawn in Table 1 and frequency analysis in Figure 19 were captured by underwater video camera clips for a few minutes each. The camera was located in one set of trials associated with Figure 19 in a fixed position, but also “roved” the tank for a general understanding of the effects of the use of the fluidic oscillation presence or absence, feedback loop length, and volumetric flowrate on the flow regime and regularity or intermittency of plumes, and the level of heterogeneity due to flow distribution in the pipework network and tank.

§4.1 Visualization studies

To investigate the impact of oscillations on the bubble size produced by membrane diffusers, an underwater video study was performed. A digital video camera recorder (Sony DCR-SR55) was secured within an underwater camera case (Sony sports pack SPK-HCD), which can withstand up to 5m of water. A high intensity discharge dive lighting system (Underwater Kinetics – Light Cannon 100 HID) was used to illuminate the recording area. The visualization system attached to a steel shaft was immersed in the water and fixed to a tank wall to carry out the recording. Two parameters were changed during the experiment. The flow rate was changed from 10 m$^3$/hr to 30 m$^3$/hr with 5 m$^3$/hr steps and the length of the feedback loop was varied from 3m to 25m. The control experiment, which involved a T-splitter instead of the oscillator, was also visualized with all above flow rates. Each operating condition change was marked with a noise signal in the video to distinguish the experiment during analysis.
Figure 21. Underwater visualization of bubble generation process at different operating regimes of the fluidic oscillator. Q, L and P represents flow rate (m³/hr), length of the feedback loop (m) and inlet gas pressure (a) Membrane at rest – blower is turned off (b) Inflated membrane with a slight pressure, but no flow (c) Control experiment with a T-junction at Q=25 (d) With fluidic oscillator, Q=25 and L=3 (e) With fluidic oscillator, Q=25 and L=15 (f) With fluidic oscillator, Q=19 and L=3 (g) With fluidic oscillator, Q=25 and L=25 - during the first half of the oscillation cycle (h) With fluidic oscillator, Q=25 and L=25 - during the second half of the oscillation cycle.
Initially with no pressure applied, the membrane stayed flat (fig. 21(a)). As the pressure was gradually increased, the flexible membrane started to inflate as shown in figure 21(b). However, bubbles were not produced until the pressure was above a certain threshold. When the pressure was further increased beyond this threshold pressure, first few bubbles started to appear from the mid section of the membrane. The membrane area of bubble production expands as the flow rate increases. Nevertheless, bubbles produced beyond a flow rate of 30 m$^3$/hr were comparatively larger; hence our analysis is focused on the best feasible flow rate.

Figure 21(c) shows a cloud of bubbles produced without oscillation at a flow rate of 25 m$^3$/hr, i.e. our control experiment with a T-junction to split the flow. The main gas flow from the blower supply two sets of diffusers having 8 units each. Therefore, the flow through each diffuse is Q/16. In this case, bubbles emerging from the membrane were estimated to be in the order of 1mm. However, an accurate size distribution analysis was not possible due to larger exposure times of the video camera.

Figure 21 (d) to (h) shows the bubble production with fluidic oscillation at 25 m$^3$/hr, but with different feed back loop lengths. With longer feed back loops, frequency is lower and the bubble stream exhibits bands of bubbles and liquid alternatively. This is clearly visible in figure 21 (g) and (h) where a 25m long feedback loop was used to oscillate the flow. In this case, as the plug of bubbles rise, liquid from surroundings flow towards the centre to fill gaps created by the rising bubbles. According to figure 21(h), some bubbles raised at the edge of the diffusers were entrained with this radial water flow causing a wavy edge to the plume. This nature of flow could also lead to coalescence of bubbles, which would eventually reduce the aeration efficiency.

From these figures it is evident that, higher frequencies eliminate the band structure of the plume; hence shorter feed back loops are preferred. Bubbles generated by a 3m feed back loop were estimated to be sub millimetre level. However, a direct comparison of bubble sizes produced by fluidic oscillation and the control experiment was not performed, as quantitative measurements were not possible. Bubble videos of the shortest feed back loop we experimented with are not available as we did not know which range to operate initially. Based on the trend we observed from above images, it follows that we can expect bubbles to be smaller with a 0.4m feedback loop. This assumption can be verified by dissolved oxygen profiles, as it performed the fastest oxygen transfer.

§4.2 Dissolved oxygen dynamics studies with microbubble aeration

It is commonly understood that aeration from dispersed bubbles is controlled by interfacial transport, and the chemical engineering mass transfer coefficient paradigm is typically adopted. The driving force for mass transfer, however, is not uniquely identified, as there are three concentrations of dissolved oxygen that control the process locally, near the bubble’s interface. The concentration in the bulk, c (mg/l), must be smaller than the saturation concentration $c^*$ at that temperature, for there to be any possibility of mass transfer.
However, the concentration of oxygen in the bubble must be greater than that dissolved in the liquid for transfer to occur. This gaseous phase concentration is rarely ever considered, as it is presumed to be a reservoir with unchanging concentration of oxygen. Clearly, however, if the volume of the microbubbles is sufficiently reduced, and thus the surface area per unit volume, and the overall area of the gaseous phase $a$, are sufficiently enhanced, the slower rise times will lead to eventual depletion of the mass transfer driving force – the concentration of oxygen in the bubble will not exceed that in the liquid phase. This suggests a natural “diminishing return” for the miniaturization of the bubble – the faster transfer coefficient factor $K_{la}$ is compensated for by depletion of the driving force. This argument was presented in Part 1 and follows from experimentation, data analysis, and modelling of mass transfer by microbubbles by Worden and co-workers. Bredwell and Worden (1997) inferred $K_l$ in an oxygen microbubble column from a plug flow concentration model for the dissolved oxygen. A laser diffraction technique was used to compute the interfacial area per unit
volume. Worden and Bredwell (1997) demonstrate that the very high mass transfer rates of microbubbles is intrinsically transient in nature. Particularly, they found that the finite capacity of oxygen in the microbubbles leads to mass transfer limitations.

Figure 22 shows the comparison between the level of measured dissolved oxygen concentration between the control and some of the best performing fluidic oscillator configurations at 25 m$^3$/h air throughput rates. The dissimilarity at long times is due to different ambient temperatures for the experiments, which were conducted outdoors so that temperature was uncontrolled, and therefore saturation DO values $c^*$ were different. These experiments were run from dissolved oxygen percentage of saturation from 17.6% to 76%, so the absolute concentrations for start and stop were mildly different.

Applying the ACSE standard to Figure 22 leads to the conclusion that all the trials are essentially the same – the difference in $K_La$ is minor. Figure 23 shows the log scale analysis of the control from which the slope after 40s yields

$$K_La = 9.212 \times 10^{-4} s^{-1}$$

Similar values are found for the trials with fluidic oscillation. Nevertheless, it is clear that the oscillators have performed better, which can be seen more clearly in Figures 24 and 25. Figures 24 and 25 are simplified to show only the best performing oscillator configuration (L=0.4m feedback loop). The curves are remarkably similar for the majority of the time histories for both the 25 and 30 m$^3$/h air throughput rates.

One key point is that the ASCE standard seeks the log-linear range and discards the initial transient before it has been achieved and anything before 20% and after 80% of saturation. For practical purposes, it should be noted that most wastewater aeration plants operate in the range of 10-30% of saturation, which actually means that the wastewater experiences the rise from 0%, assuming the incoming mixed liquor has depleted all available oxygen, to this level. Therefore the ASCE standard only operates slightly within the practically important range. If microbubbles are being generated, then the deeper analysis of Worden and co-workers explains that the “dovetailing” of all of these curves in Figure 22 is predicated on the exhaustion of the mass transfer driving force because of the finite capacity oxygen content of the microbubbles. Since the mass transfer is so rapid, eventually the depleted oxygen partial pressure in the bubble reaches its saturation limit, and it does this faster than larger bubbles. Although this is a drawback for microbubble operation, it is not practically important as it is levels of DO greater than 40% where this limitation becomes apparent.

The better performance of microbubbles is in the early “transient” stage, which given the analysis of Worden and co-workers, always applies to microbubbles. There is not a clear “log-linear” regime, although you can fit a log-linear expression to the curves, as the regression is a well-posed inverse problem with a unique solution. But throwing away the early transient stage in the regime 0-30% of saturation brings to mind the adage of “throwing away the baby with the bath water”. It is in the initial transient stage that microbubble aeration driven by fluidic oscillation significantly outperforms steady flow aeration, so it is necessary to develop measures of this performance, which is done later in this section. This is the regime of practical importance as it maps to “residence time” in steady operation of a wastewater aeration system.

Further support for the better performance of the oscillator driven trials comes from the conduct of the trials by way of information noticed in passing. After each trial run, the tank was deoxygenated by dosing, through the same oscillator-pipework system, nitrogen gas,
which forces out the oxygen from the water. For the first experiment, we used the control configuration, and thereafter the best performing configuration on the aeration experiments to deoxygenate with nitrogen. The control took 3 cylinders of nitrogen to deoxygenate from 76% of saturation to 17%. Eventually, the best performing deoxygenation was the L=0.4m feedback loop, which took one and a half cylinders to achieve. It should be noted that the control was replicated at least three times for 25 m$^3$/h air throughput rate, as it has been mooted that the first experiment would suffer from any residual biological oxygen demand in the tank. The replicants were very similar in profile to Figures 22/23. The first experiment was with the L=6m feedback loop, and its replicant did show a modest improvement in DO profile.
The long time similarity ("dovetailing") is not apparent, however, during the initial stage of the experiments. There is an initial burst in the oscillator dissolved oxygen levels in both experiments, which is not evident in the control, which is flat for a long while, at least 150 seconds. We conducted repeat trials to determine for both the control and the fluidic oscillator configuration whether or not the features of the "delay time" for the control and the sharp acceleration of the oscillated aeration were reproducible, and they were. Other oscillation frequencies experiments were conducted and are reported on in the internal report. In general, there were two ranges of frequency for which this high initial aeration acceleration were observed – the fastest frequencies studied (L=0.4m and 2m feedback loops) and the slowest (L=25m feedback loop). From the visualization studies, it was apparent that the slow frequency oscillation set up a large coherent toroidal structure emanating from the diffuser, which we dubbed the "jellyfish" mode. We did not observe the fast frequency response at L=0.4m feedback loop as it was suggested by preliminary aeration studies results that as the frequency increased, the initial acceleration did as well.

Since we did not observe that fastest frequency flow response, we can only speculate that we were exciting a resonant mode of the diffuser membrane slit, most likely a harmonic, and therefore slicing off smaller bubbles. The total surface area factor “a” in the mass transfer coefficient factor $K_La$ would therefore be increased, leading to more rapid initial mass transfer when the driving force is the difference between the gaseous oxygen concentration and the liquid DO level. However, these smaller bubbles would deplete the bubble oxygen content faster, and therefore as the DO levels rise, would gradually lose their advantage over larger bubbles with greater retained oxygen levels. Thus the long time similarity as the surface area factor $a$ no longer dominates the transfer process, and the diminished driving force becomes limiting.

How can we quantify this initial aeration acceleration? Firstly, we must argue that it is important. In industrial wastewater treatment processing, the dissolved oxygen levels are not so high that the saturation limitation nor the depletion of microbubble oxygen content is likely to be important.

Typically, aeration plants operate at 1-3 mg/l DO, for which the process is typically contacting unaerated or low level DO mixed liquor with aeration plumes from the diffusers. So the short residence time region of the aeration trials is relevant for industrial practise, and the later, similar behaviour is not.

The traditional analysis of aeration transient data sets is to use a fitting curve which is argued to be appropriate:

$$c = c^* + (c_0 - c^*) e^{-K_La t}$$ (17)

It should be stressed that this fitting curve is not derivable from any bubble mass transfer model, and in particular does not respect that there are two driving forces – difference between liquid DO and bubble gaseous phase oxygen concentration and between liquid DO and saturation value. This sigmoidal shape curve only respects the latter mechanism operates. In the internal report, we applied the standard ASCE methodology for fitting (17) to all our data sets for the clear water trials. The methodology is to discard both initial data and the approach to saturation, plot the data on a log-linear scale, and report the negative of the slope, which is then “identified” as $K_La$. According to this methodology, all our $K_La$ values are practically identical, since the initial acceleration, which we have argued is the
dominant feature of the oscillator technology, is discarded. Clearly, it cannot capture this initial acceleration effect.

In this subsection, we introduce two concepts for quantifying the initial acceleration in aeration, and a third is introduced in the next subsection. The first concept is the time needed to achieve a certain level of DO, where the level is arbitrary, but since DO=3mg/l is an important threshold for industrial practice, it is chosen. Table 2 tabulates this quantity for our clear water trials varying with three different air flow rates (16, 25, 30 m$^3$/h). Not surprisingly, the control shows a trend that increasing throughput of air decreases the time needed to rise to 3mg/l. At the lowest throughput achievable without stalling the blowers, 16 m$^3$/h, the rise time to 3mg/l is lowest for the control configuration. At higher throughput rates (25, 30 m$^3$/h), however, the oscillatory configurations all rise to 3mg/l faster.

Table 2 Clear Water: Time (s) to achieve 3 mg/l

<table>
<thead>
<tr>
<th>Flow rate (m$^3$/h)</th>
<th>Control</th>
<th>l=0.4m</th>
<th>l=2m</th>
<th>l=3m</th>
<th>l=25m</th>
</tr>
</thead>
<tbody>
<tr>
<td>16</td>
<td>290</td>
<td>336</td>
<td></td>
<td></td>
<td>415</td>
</tr>
<tr>
<td>25</td>
<td>238</td>
<td>126</td>
<td>186</td>
<td>148</td>
<td></td>
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<tr>
<td>30</td>
<td>220</td>
<td>166</td>
<td>177</td>
<td>187</td>
<td>173</td>
</tr>
</tbody>
</table>

Table 3 Clear Water: Initial slope (mg l$^{-1}$ s$^{-1}$)

<table>
<thead>
<tr>
<th>Flow rate (m$^3$/h)</th>
<th>Control</th>
<th>l=0.4m</th>
<th>l=2m</th>
<th>l=3m</th>
<th>l=25m</th>
</tr>
</thead>
<tbody>
<tr>
<td>16</td>
<td>0.00144</td>
<td>-0.00128</td>
<td></td>
<td></td>
<td>-0.0113</td>
</tr>
<tr>
<td>25</td>
<td>-0.00194</td>
<td>0.0262</td>
<td>0.0142</td>
<td>-0.0139</td>
<td></td>
</tr>
<tr>
<td>30</td>
<td>0.00139</td>
<td>-0.01211</td>
<td>-0.0065</td>
<td>-0.00483</td>
<td>-0.0065</td>
</tr>
</tbody>
</table>

In Table 2, the achievement time for 3mg/l is computed by cubic spline interpolation and Newton’s method for root finding. The best rise time is given by the fastest frequency achievable for the L=0.4m at 25 m$^3$/h, roughly half that of the control.

The second figure of merit for the initial acceleration is the initial slope of the DO level curve. If formula (17) represents the interfacial transport dynamics faithfully, then the initial slope is given by

$$\frac{dc}{dt}_{t=0} = (c^* - c_0)K_La$$

(18)

Thus, since $c^*$ and $c_0$ are known, $K_La$ can be estimated from the initial slope. Given that for our experiments, the factor $(c^* - c_0) = 8$mg/l is a near constant for the trials, the derivative estimated at $t=0$ is a sufficient indicator of the trend in $K_La$. Table 3 shows the estimates of the initial slope in dissolved oxygen estimated from the cubic spline fit of the curve and extrapolated to $t=0$. Generally, the level of the initial slopes at $10^{-3}$ mg l$^{-1}$ s$^{-1}$ suggest that initial slopes are fairly flat, and may even dip due to inhomogeneities in the tank being mixed past the sensor. If $10^{-3}$ is the level of the “noise” in the experiment if the initial slope is flat, however, it is clear that the initial slopes that are an order of magnitude larger are significant and not “flat”. Indeed, the best performing initial acceleration occurs with the fastest frequency achievable for the L=0.4m at 25 m$^3$/h.
Given that nearly all the measurements are initially noise around “flat”, the initial slope is probably not a good figure of merit, even though it also identifies the same best performing configuration. In the next subsection, we develop a measure of initial exposure which is less influenced by the fluctuations in the DO level than the initial slope, but does capture the essence of oscillatory acceleration.

**Aeration dosage as an indicator of biological activity**

Dosage is frequently used in field studies of atmospheric and oceanographic turbulent dispersion (Zimmerman and Chatwin, 1995a,b). Dosage, in its simplest form, is defined in an interval of time of duration $T$ as:

$$D = \frac{1}{T} \int_{0}^{T} C \, dt$$  \hspace{1cm} (19)

where $C$ is the concentration of the key chemical species. Clearly this is equivalent to the average concentration over the time interval. Many toxicology studies have shown a correlation between the dosage, rather than peak concentrations, between exposure and toxicological response. Intuitively, the dosage is the availability in both amount and duration of the chemical species within a volume that the bioculture can make use of metabolically. In this section, we will consider the dosage in the first three minutes of both the clear water and mixed liquor aeration experiments, with $C$ taken as the dissolved oxygen concentration. We take $T=180$ s as dissolved oxygen levels between 1 and 3 mg/l were prevalent in these studies, and these levels are important in industrial practise.

Table 4  Clear Water: Additional dosage over first three minutes of trial (mg/l)

<table>
<thead>
<tr>
<th>Flow rate (m$^3$/h)</th>
<th>Control</th>
<th>$l=0.4m$</th>
<th>$l=2m$</th>
<th>$l=3m$</th>
<th>$l=25m$</th>
</tr>
</thead>
<tbody>
<tr>
<td>16</td>
<td>0.716</td>
<td>0.838</td>
<td></td>
<td>-0.00431</td>
<td></td>
</tr>
<tr>
<td>25</td>
<td>1.05</td>
<td>3.31</td>
<td>1.87</td>
<td>2.97</td>
<td></td>
</tr>
<tr>
<td>30</td>
<td>0.759</td>
<td>2.66</td>
<td>2.43</td>
<td>1.64</td>
<td>2.57</td>
</tr>
</tbody>
</table>

Table 4 shows the dosage for the control and at least two oscillation configurations at three aeration flow rates (16, 25, 30 m$^3$/h). The integration was conducted using the trapezoid rule. The highest dosage for a control was achieved at 25 m$^3$/h, at 1.05 mg/l. The highest dosage overall was achieved at the same flow rate with the fastest oscillation frequency achievable at the $L=0.4m$ feedback loop configuration, at 3.31mg/l. This greater than three-fold improvement, indeed more than four-fold over the design target of the control at 30 m$^3$/h, suggests that biological metabolic activity will be much higher due to the higher availability of dissolved oxygen during this period. For industrial practise, which would benefit from fast residence times in the dissolved oxygen regime between 1 and 3 mg/l, this suggests that the $L=0.4m$ feedback loop configuration is the most effective configuration of those trialled, at 83% of the design flow rate. Both of these features contribute to potential energy efficiency – designing for higher throughput, but with lower per diffuser air flow.

§4 Conclusions

In this report, we have introduced the concept of how fluidic oscillation leads to smaller bubbles in a number of scenarios, including the possibility of retrofitting the feed to a membrane diffuser network. A bespoke experimental rig was designed to achieve conventional aeration for a tank with a 30 m$^3$/hr throughput through the diffuser network. The major findings are:
A visualization study of the underwater generation of microbubbles with the control flow configuration and a range of feedback loop lengths leading to different oscillation frequencies. The study identified candidates for best aeration operating conditions.

A characterization of the oscillation frequency from acoustic analysis for different feedback loop lengths and volumetric air flow rates.

A measure of power consumption for different feedback loop lengths and volumetric air flow rates which identifies a significant reduction for oscillatory flow over steady flow, with maximum value of 18% reduction at the best aeration configuration.

A theoretical discussion of the origin of the decreased power consumption.

A clear water dissolved oxygen study that identifies that for some oscillatory configurations, there is a significant initial acceleration in aeration rate. The time for achieving 3mg/l level of DO concentration was nearly half that of the control, and the dosage exposure 3-4 fold better, for the best oscillatory configuration performance, which was achieved at 83% of the design volumetric flow rate.

Clearly, these results are promising that in the practically important regime for wastewater aeration, fluidic oscillation can be introduced by an above water change in the piping configuration to introduce a jet diversion fluidic oscillator or oscillators with the potential for both energy savings and improved aeration performance. Whether or not this is achievable should be assessed by near equipment scale investigations of mixed liquor aeration as a controlled scientific experiment to compute the alpha factors, if applicable, but otherwise the transient effects such as dosage and time to achieve certain DO levels, as well as the overall performance in removal of COD, BOD, and TOC. Hwang and Stenstrom (1985) and Tchobanoglos and Burton (1992) detail such trials.

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