An integrated study of parallel valveless micropumps

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Abstract We describe an analytical, computational and experimental study of parallel valveless micropumps. A one dimensional model of a parallel micropump is presented and compared with available experimental data. The model confirms the linear decrease of the volume flux with pressure rise which is consistent with the experiments. The computational study showed a similar linear decrease but highlighted the effect of turbulence closures on the rectified mean flow, with the experimental data sitting between the turbulent and laminar closure regimes. The experimental study confirmed the importance of the displacement distance of fluid through the nozzle compared to nozzle length in the setting whether the flow regime is streaming or rectified. General conclusions are made about how to improve the pumping efficiency of micropumps.

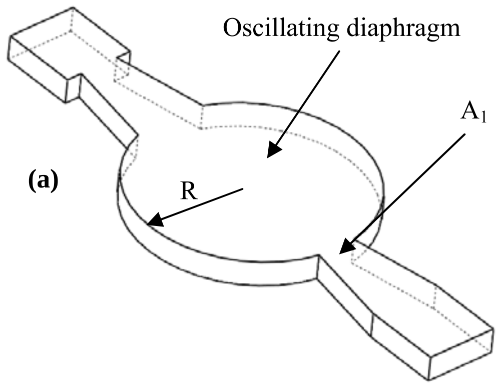
Keywords: CFD, Valveless, Micropumps, Analytical

1. Introduction

During last two decades there have been several attempts to design and construct valveless micropumps (Geipel et al. 2007, Gerlach, Wurmus 1995, Gerlach, Wurmus 1996, Goldschmidtboing et al. 2006, Inman et al. 2007, Izzo et al. 2007, Olsson et al. 1995, Stemme, Stemme 1993, Ullmann, Fono 2002, Ullmann 1998) over a wide range of length-scales (from 40mm to 10mm in chamber diameter), constructed from polymers, brass, glass and etched silicon. In most of these studies, the pumping efficiency is determined experimentally. The lack of understanding of the complex fluid structure interactions that occur in valveless pumps currently impedes progress in this area.

One of the leading candidates for the application of microfluidic systems is the valveless rectification micropump, largely due to the ease of manufacture. This type does not have moving valves and instead contains diffuser/nozzle elements as rectifying agents which make them less susceptible to clogging. Figure 1a shows a typical single-chamber valveless micropump while figure 1b shows a typical parallel double-chamber valveless micropump (PDVM).

(a)



(b)

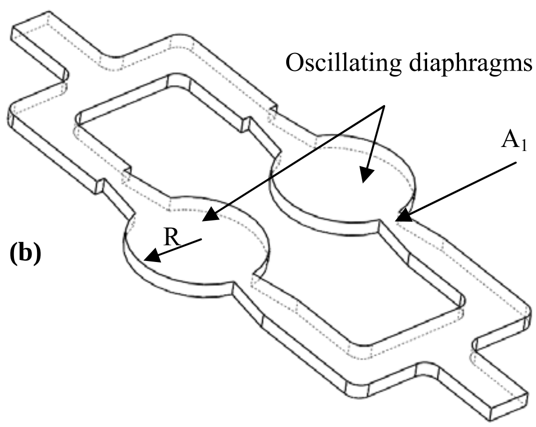


Figure 1 Schematic of valveless micropumps (a) single chamber (b) double-chamber parallel.

In this paper we study the physics of PDVM using an analytical theory (extending the analysis of Eames et al. (2009)), new high resolution computations and experiments. This unified approach provides a clearer view of the critical processes which are important for achieving both high mean flow rates and pressure rise.

2. Analytical Model

To understand the underlying physics and parameters that affect the performance of valveless micropumps a robust mathematical model is required. There have been several attempts to model valveless micropumps mathematically (Gerlach, Wurmus 1995, Pan et al. 2001, Pan et al. 2003, Stemme, Stemme 1993, Ullmann, Fono 2002, Ullmann 1998) to estimate the performance of this type of micropumps. The main problem with most formulations is that they do not give closed form expressions (Gerlach, Wurmus 1995, Pan, Ng, Liu, Lam, Jiang 2001, Pan, Ng, Wu, Lee 2003, Ullmann, Fono 2002, Ullmann 1998) that can be adapted and applied to other situations. The most relevant to our discussion is the model of Olsson et al. (1995) and Stemme, Stemme (1993), described later, which is based on a quasi-steady analysis.

Recently Eames et al. (2009) developed a one-dimensional model which considered fluid dynamics features of one-chamber valveless micropumps with some simplifications. This model presents an estimation of the maximum volume flux and maximum back pressure of a one-chamber valveless micropump in a closed form. This model agreed well with the published experimental results of Olsson et al. (1995) (see Azarbadegan et al. (2009)).

We extended Eames et al. (2009) model further to estimate the performance of double-chamber parallel micropumps. Based on Eames et al. (2009) model for one-chamber valveless micropumps the maximum volume flux (Q_0) and the maximum back pressure (ΔP_{max}) when the frictional losses are insignificant can be estimated as follows:

$$Q_0 = \frac{\pi V_m \omega}{16}\frac{\zeta_- - \zeta_+}{\zeta_- + \zeta_+}, \tag{1a}$$

$$\Delta P_{max} = \frac{\rho V_m^2 \omega^2}{16 A_1^2}(\zeta_- - \zeta_+), \tag{1b}$$

whereas it can be seen in figure 1 for a chamber radius of R, forcing amplitude of X_d, and angular frequency of ω, the chamber volume V_c oscillates with amplitude V_m, described by $V_m = \alpha\pi R^2 X_d$, where α is a dimensionless coefficient which characterises the shape of the deformed surface. In addition, ζ_+ and ζ_- are the pressure-drop coefficients of the diffuser and nozzle respectively, and A_1 is the area of the throat of diffuser/nozzle elements.

This model can be adapted and applied to estimate the performance of a PDVM. The maximum back pressure (ΔP_{max}) for this type of micropump is the same as a single-chamber micropump because both chambers are in parallel. In the case of insignificant frictional losses the maximum volume flux will be doubled. It should be noted that in this model the effect of phase difference for the actuation forces of two chambers are not considered. Therefore, based on the discussion above, the maximum volume flux (Q_0) and the maximum back pressure (ΔP_{max}) for a double-chamber parallel micropump when the frictional losses are insignificant are

$$Q_0 = \frac{\pi V_m \omega}{8}\frac{\zeta_- - \zeta_+}{\zeta_- + \zeta_+}, \tag{2a}$$

$$\Delta P_{max} = \frac{\rho V_m^2 \omega^2}{16 A_1^2}(\zeta_- - \zeta_+). \tag{2b}$$

3. Computational Study

One way to understand the characteristics of valveless micropumps is using computer simulation. However, there are some limitations for simulating these devices. To simulate micropumps robustly a multi-physic model is required. In the simplest form, simulation of the oscillating diaphragm and its interaction with the fluid is desirable which requires fluid-structure interaction (FSI) capabilities. FSI simulation is computationally expensive and it takes a long time to simulate several models. One way to simplify the computational model is to use computational fluid dynamics (CFD) models only with mesh deformation.

In this paper we used the latter approach to simulate a PDVM. One of the main limitations in this simplified model is defining the realistic boundary conditions for the problem, in other words, as the flow is pulsatile both pressure and velocity are functions of time at inlet and outlet boundaries.

Olsson et al. (1995) designed and studied a PDVM with flat-walled diffuser/nozzle elements. The two identical pump chambers had radii of R = 6.5 mm and depth of 0.3 mm. The flat chamber walls were driven by four piezoelectric discs of radii 5 mm. The diffuser/nozzle elements had a throat width of 0.3 mm, an outlet width of 1.0 mm, and length of 4.1 mm with a slightly rounded inlet. When the piezoelectric actuators were driven in anti-phase mode the excitation frequency was 540 Hz. We used Olsson et al. (1995) device to define our computational model in the CFD analysis. A commercial package (ANSYS CFX) is used to simulate this PDVM.

The geometry of the computational domain is shown in figure 2 .The oscillating diaphragms are modeled as moving boundaries and the inlet/outlet boundary conditions are modeled as openings (i.e. fluid can go in both directions) with pressure boundary conditions.

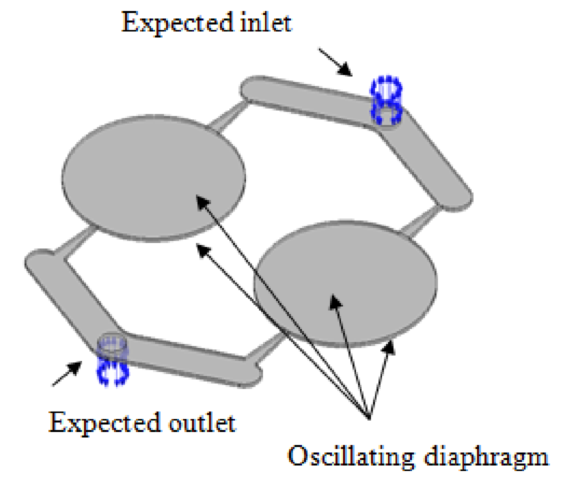


Figure 2 Computational domain and boundary conditions for CFD analysis

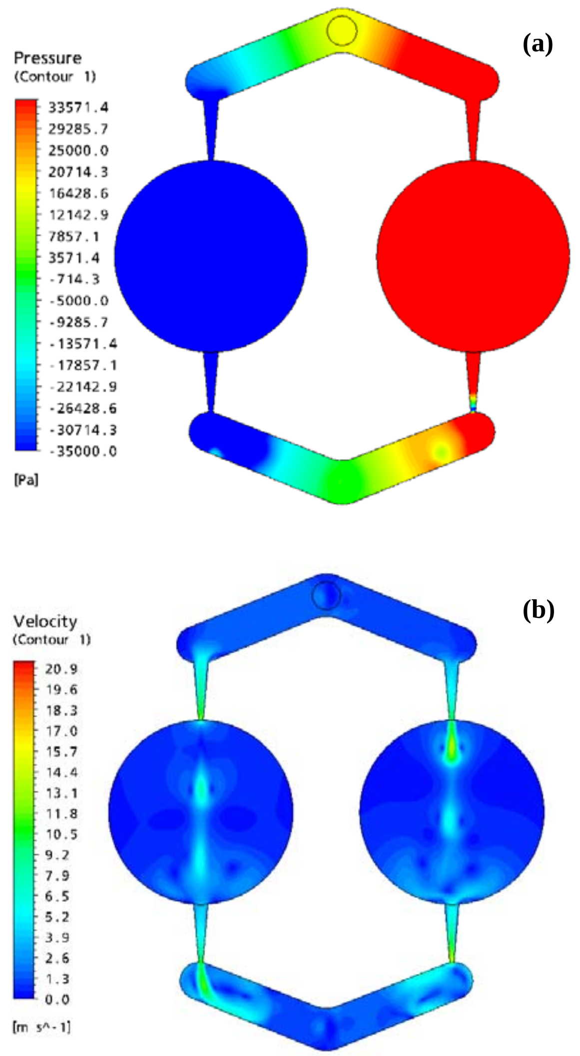


Figure 3 Pressure contours (a) and velocity contours (b) for a time step in which the right chamber is in suction mode and the left chamber is in pumping mode

The maximum diaphragm displacements are specified in Olsson et al. (1995). Based on these values the deformation of the diaphragms are modeled as paraboloids which corresponds to $\alpha = 0.5$. Velocity and pressure contours for a time step are shown in figure 3. In the corresponding time step the right chamber is in suction mode while the left chamber is in pumping mode.

4. Experimental Study

There have been many published experimental studies of single chamber valveless micropumps and a few multi-chamber micropumps. The most comprehensive study of PDVM is Olsson et al. (1995).

Our experimental study involved a novel use of micromachining to general large pumps, as shown in figure 4. The chamber diameter is 40 mm, and it is milled from PMMA. The advantage of using PMMA is that the chamber and channels can be accessed optically. The chambers are driven by a piezoelectric actuator at a range of frequencies (1-27 kHz) and voltages (1-130 V).

The cross-sectional area of the diffuser/nozzle elements plays a strong role in setting up a rectified mean flow. Our experiments initially showed that for (~3 mm) wide nozzles, a streaming flow was setup (consistent with the analysis of Eames et al. (2009)) because the volumetric displacement caused by the piezoelectric device is small. Further experimental refinement using narrower (~1 mm) nozzles generated a rectified mean flow over a wide range of driving frequencies.

5. Results and Discussion

As a starting point we make a comparison between the analytical model and the experimental results of Olsson et al. (1995). This is shown in figures 5(a) and 5(b). The information used to make the comparison is given by Olsson et al. (1995) , including the diffuser/nozzle element characteristics; α was estimated to be 0.5 (for a simply supported plate). The predictions are sensitive to α and ζ_-, ζ_+, but the similar trend, such as the linear dependence of the flow rate on pressure rise is captured. The predicted flow rate is comparable to the measured values, but the pressure rise is overestimated.

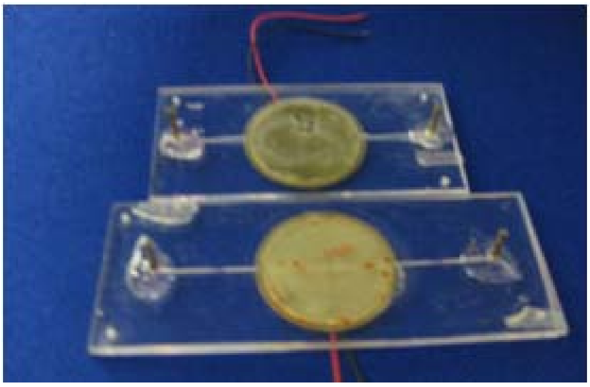


Figure 4 Two single-chamber valveless micropumps fabricated by means of micromilling of PMMA

The results of a computational study are shown for comparison in figure 5(c). As the Reynolds number in the domain varies over the range 0−4600, the flow may either be laminar or turbulent (k-ε) regimes. Figure 5(c) shows a comparison between experimental data and simulation results based on using either a turbulent or laminar closure. While the slope in both curves are the same (and consistent with the experiments and analytical model), both the gradient and intercept with the axes are different with that of the experiments.

For the highest flow rate, the turbulent closure model would be the most appropriate, while for the highest pressure rise, when the mean flow is low, the laminar flow model would be the most appropriate. This appears to be consistent with the experimental results, which sit at an intermediate position between both curves. This underlines the difficulty of applying general codes (to account for turbulence / laminar flows) to this class of problems.

5. Conclusions

The broad conclusions from this three-fold study using analytical, computational and experimental approaches are:

a) The displacement of the chamber diaphragm compared to the cross-sectional area of the diffuser/nozzle throat controls whether the flow regime is streaming or rectified. Rectification is improved by decreasing the throat cross-sectional area.
b) Both pumping efficiency and flow rate depend on the shape of the diffuser/nozzle element. Pumping efficiency is determined by the difference of the pressure loss coefficients $(\zeta_{-} - \zeta_{+})$, while the flow rate is determined by the ratio (ζ_{-}/ζ_{+}).
c) Computational aspects highlight the importance of (i) resolution and (ii) boundary conditions. The transitional nature of the flow (and the high Reynolds number associated with the narrow diffuser/nozzle throat) means the flow is turbulent (Re_max>4600), pulsatile and must be resolved to calculate accurately Q_0 and ΔP_{max}. The boundary conditions for PDVM flow depend on whether the circuit is open or closed. The challenge with open circuits is that the inlet and outlet change depending on the phase of the forcing of the pump chamber.
d) The experiments show that pumping efficiency is enhanced close to the natural frequency of the system. But at the natural frequency the entire system vibrates which is detrimental from a structural point of view.

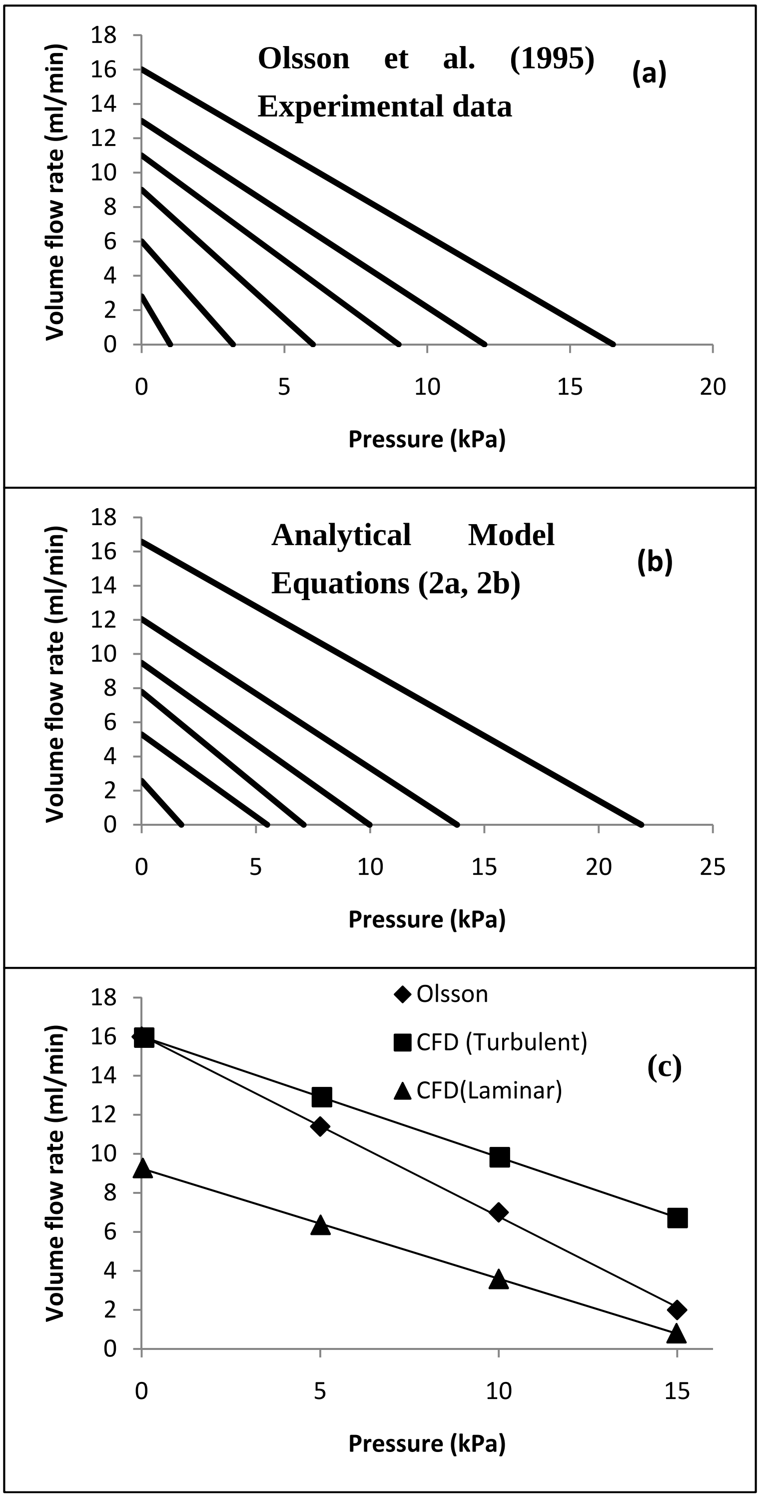


Figure 5 Performance charts based on (a) experimental data from Olsson et al. (1995) and (b) one-dimensional model (equation 2a, b), and (c) comparison between experimental data and computational results

Acknowledgments

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