ARBITRARY LAGRANGIAN EULERIAN ANALYSIS OF A BIDIRECTIONAL MICRO-PUMP USING SPECTRAL ELEMENTS

ALI BESKOK
Mechanical Engineering Department, Texas A&M University, College Station, TX-77843, USA
E-mail: abeskok@mengr.tamu.edu

TIMOTHY C. WARBURTON
Division of Applied Mathematics, Brown University, Providence, RI 02912, USA
E-mail: timw@cfm.brown.edu

Performance analysis of a reversible micro-pump system is obtained by numerical simulations. The unsteady incompressible Navier-Stokes equations are solved in a moving micro-pump system using a spectral h/p element algorithm, employing an arbitrary Lagrangian Eulerian (ALE) formulation on structured/unstructured meshes. The performance of the micro-pump is evaluated as a function of the Reynolds number and the geometric parameters. The volumetric flowrate is shown to increase as a function of the Reynolds number. The unsteady traction forces on the pump membrane and the vorticity dynamics within the pump cavity are presented.

Keywords: Bidirectional Micro-Pumps; CFD for MEMS; ALE; Spectral H/P Methods.

1. Introduction

Micro-pump systems delivering volumetric flow rates in the order of $10^{-8} \sim 10^{-12}$ m$^3$/s can be used in various biofluidic, drug delivery, mixing and flow control applications. Most of the micro-pump systems are actuated by a vibrating membrane in a chamber with hanging-beam-type (Cantilever beam) inlet and exit micro-valves.$^{1-3}$ Since the Cantilever-type micro-valves only open in a preferred flow direction, designs utilizing such valves are strictly unidirectional. Bidirectional micro-pumps have also been proposed, where a rotating cylinder is located asymmetrically within a micro-channel, and it propels the fluid due to the viscous action while turning with a prescribed angular speed.$^{4,5}$ This “novel micro-pump” works well for low Reynolds number flows. However, its efficiency rapidly diminishes with increased Reynolds number, as fluid inertia takes over. Also for gas flows the small dimensions of this device, required to maintain small Reynolds numbers, create a further complication. When the local Knudsen number defined as the ratio of mean freepath $\lambda$ to the channel width $h$, approaches $Kn \approx 0.01$ the “velocity slip effects”
start to be important. The performance for this shear-driven flow pump decreases with the increased slip effects. The mean free path of air $\lambda$ at atmospheric conditions is 0.65 nm. Therefore, the slip effects start to become important, when the characteristic dimensions ($h$) approach $5 \mu m$. A detailed analysis of momentum and energy transport in micro-scales has been presented in Refs. 6 and 7.

In this study, we present a bidirectional (reversible) micro-pump utilizing a vibrating membrane and piston-type (moving) inlet and exit valves (See Fig. 1). Since the inlet and the exit valves are micro-pistons oscillating between the open and closed positions with a prescribed motion, it is possible to control the performance of the micro-pump with these valves. The design is flexible, such that the pumping direction can be reversed, and performance degradation is not observed for increased Reynolds numbers.

The concept verification and analysis of the micro-pump is achieved by numerical simulations, demonstrating the possibility of using CFD simulation tools for computer aided design of Micro Electro Mechanical Systems (MEMS). This paper is organized as follows: in the next section we present the design parameters of the micro-pump. In Sec. 3, theoretical analysis of the pump performance is presented. This is followed by the description of the numerical algorithm in Sec. 4, and performance predictions with numerical simulations in Sec. 5. Finally, we summarize our results in Sec. 6 and discuss future applications.

2. Geometric Specifications

The micro-pump geometry is presented in terms of the length of the membrane “$L$”, in Fig. 1. This allows us to interpret the results using the dynamic similarity concept in determining the optimum pump dimensions and operation conditions.

The micro-pump is ideally placed between two reservoirs. However, numerical simulation of such a system is difficult due to large reservoir size. Any finite size reservoir would require inflow and outflow numerical boundary conditions, and these

![Fig. 1. Sketch of the micro-pump operating between two micro-channel systems. Inlet and exit valves open and close periodically with maximum gap of $g_{\text{max}} = 0.125L$ and minimum gap of $g_{\text{min}} = 0.025L$.](image)
must be imposed carefully in order to avoid a preferred flow direction in the simulations. This difficulty is overcome by placing the micro-pump between two symmetric micro-channel flow systems, where equal amount of liquid flow is maintained from top to bottom direction. Therefore, the flow conditions are symmetric and there is zero net flow from one channel to another when the pump is not actuated. We verified this by numerical simulations.

Oscillation of the membrane with a specified frequency \( \omega \) and an amplitude \( a \) excites the fluid within the micro-pump cavity. For our simulations we have used \( a = L/10 \), and \( \omega = \pi c/L \), where \( c^2 = T/M \), \( T \) being the uniform tension, and \( M \) being the mass per unit area of the membrane. The first-mode of vibration of the membrane is used to determine the deflection (See Fig. 2, top), velocity and the acceleration of the membrane as a function of time,

\[
\begin{align*}
y(x,t) &= a \sin \frac{\pi x}{L} \sin \frac{\pi \omega t}{L}, \\
v(x,t) &= a \left( \frac{\pi c}{L} \right) \sin \frac{\pi x}{L} \cos \frac{\pi \omega t}{L}, \\
\dot{v}(x,t) &= -a \left( \frac{\pi c}{L} \right)^2 \sin \frac{\pi x}{L} \sin \frac{\pi \omega t}{L}.
\end{align*}
\]

Since the membrane motion is imposed as a boundary condition, the numerical simulations show the response of fluid flow to the prescribed membrane motion,

![Diagram of membrane deflection and valve positions](image-url)

Fig. 2. Top: deflection of the membrane \( y(x,t) = a \sin(\pi x/L) \sin(\omega t) \), bottom: the position of the valve tips during a pump cycle.
and the simulations do not correspond to a coupled fluid/structure interaction solution.

Since, the pump geometry is symmetric, active pumping cannot be achieved solely with the membrane oscillations. Hence, we included the inlet and exit valves, which open and close periodically to break the flow symmetry. The inlet and the exit valves are located at the mean height of $y_0 = 0.325L$ from the membrane, and the position of the valves are specified as a function of time,

$$y(t) = y_0 \pm 0.05L \tanh \left[ 4 \cos \frac{\pi ct}{L} \right].$$

The valve motion is designed to be closed to a step function, oscillating between open and closed positions with finite velocity and acceleration. The gaps between the valve tips and the top wall of the micro-pump are $g_{\text{max}} = 0.125L$ and $g_{\text{min}} = 0.025L$, during the open and closed positions, respectively. We have the choice to close the valves completely so that the gap between the valve tips and the top wall are vanished. However, this would have annihilated several elements between the gap and the wall, requiring remeshing of the computational domain. For computational simplicity we avoided closing the valves completely and we left a finite gap between the valve tips and the wall at all times. This introduced “leakage” to our simulation results. The position of the inlet and exit valves during a cycle of the micro-pump is presented in Fig. 2 (bottom). The phase difference between the inlet and the exit valve is $\pi$ radians.

3. Theoretical Analysis

The performance of our design is based on the following factors: the membrane length $L$, the membrane width $W$, the pump-cavity height $H$, the amplitude of vibration of the membrane $a$, the frequency of vibrations $\omega$, the minimum valve clearance (the gap between the closed-valve and the top wall) $g_{\text{min}}$, the time-lapse between the opening and the closing of the valves (see Eq. (4), and Fig. 2, bottom) $\delta^{-1}$, dynamic viscosity of the fluid $\mu$, and fluid density $\rho$. There are nine variables associated with performance of the micro-pump, with dimensions of length, time and mass. This corresponds to six nondimensional variables: $a/L$, $W/L$, $H/L$, $g/L$, $\delta/\omega$ and $a^2\omega/\mu$. In this study, we have fixed $\delta = 0.15\omega$. The geometric length-scales are set as $H = 0.4L$, $g = 0.025L$. The parameters $W/L$ and $a/L$, and $a^2\omega/\mu$ are varied.

The magnitude of the membrane velocity $u \simeq \omega a$. Therefore the parameter $\rho a^2\omega/\mu = \rho a\omega/\mu$ is a Reynolds number. Using the kinematic viscosity the Reynolds number becomes $Re = a^2\omega/\nu$. Since the reference velocity is based on the vibration frequency of the membrane, the Strouhal number is unity for all simulations.

The volumetric flowrate per channel width ($W$) can be calculated using Eq. (2),

$$\frac{\bar{Q}}{W} = \int_0^L v(x,t)dx = -2ac \cos \frac{\pi ct}{L}.$$

(5)
The suction stage of the micro-pump happens while \(-L/2c \leq t \leq L/2c\). Therefore, the average volumetric flow for a given period \(T = \omega^{-1}\) is:

\[
\dot{Q} = \frac{2acW}{\pi} \int_{-\frac{\pi}{4}}^{\frac{\pi}{4}} \cos \left(\frac{\pi c t}{L}\right) dt = \frac{4aLW}{\pi},
\]

and the average flowrate becomes,

\[
\dot{Q} = \frac{4aLW\omega}{\pi} = \frac{4}{\pi} \frac{L}{a} W^\n Re.
\]

This simple analysis indicates that the volumetric flowrate is proportional to the Reynolds number, the width of the micro-pump membrane \(W\), and the \(L/a\) ratio. Our analysis assumes no leaks from the inlet valve during the ejection stage, and from the exit valve during the suction stage. Therefore, Eq. (7) gives the maximum theoretical volumetric flowrate of the micro-pump system. This value will be used in the next section in determining the efficiency of the micro-pump, when the leakage due to the finite gap between the valve tips and the top wall at closed valve position are considered.

4. Numerical Formulation

The arbitrary Lagrangian Eulerian (ALE) method enables the solution of fluid flow in arbitrarily moving domains, and it forms a bridge between the familiar Eulerian and Lagrangian descriptions of motion. In the limit of the grid moving with the fluid particles the Lagrangian description is obtained, and if the grid is stationary the formulation is Eulerian. However, it is also possible to assign the grid motion independently of the fluid flow, hence the ALE method can be used for simulation of fluid flow and transport in moving domains, especially in fluid/structure interaction problems and in numerical simulation of thermal/fluidic transport in Micro Electro Mechanical Systems (MEMS).

The ALE method has been developed by various researchers. However, the first application of an ALE algorithm to quadrilateral spectral elements was achieved by Ho. In this paper we utilize a mixed structured/unstructured spectral h/p element algorithm \(N\varepsilon\kappaT\alpha\varepsilon\ ALE\), which is an extension of the unstructured spectral element methodology developed in Refs. 17 and 18. The advantages of using the ALE formulation with spectral element method are:

1. High order accuracy of the h/p scheme with minimal dispersion and diffusion errors.
2. Spectral elements are larger and less numerous than finite elements or finite volume cells, hence they can support larger deformations without becoming entangled.
3. The triangular elements can support deformation without losing excessive resolution.
The nondimensionalized incompressible Navier–Stokes equations in a time-dependent domain \(\Omega(t)\) moving with velocity \(w\) is given by:

\[
\frac{\partial \mathbf{u}}{\partial t} + (\mathbf{u} - w) \cdot \nabla \mathbf{u} = -\nabla p + \frac{1}{Re} \nabla^2 \mathbf{u} + \mathbf{F} \quad \text{in } \Omega(t),
\]

\[
\nabla \cdot \mathbf{u} = 0 \quad \text{in } \Omega(t),
\]

where \(Re\) is the Reynolds number and \(\mathbf{F}\) is a body force. The time discretization is based on the third-order accurate (in time) stiffly stable operator splitting scheme developed by Karniadakis et al., and it involves the following steps,

\[
\mathbf{u}^{n+1} = \sum_{q=0}^{J_i-1} \alpha_q \mathbf{u}^{n-q} + \Delta t \left( \sum_{q=0}^{J_e-1} \beta_q N(\mathbf{u}^{n-q}, \mathbf{w}^{n-q}) + F^{n+1} \right),
\]

\[
x^{n+1} = \sum_{q=0}^{J_i-1} \alpha_q x^{n-q} + \Delta t \left( \sum_{q=0}^{J_e-1} \beta_q w^{n-q} \right),
\]

\[
\frac{\partial p^{n+1}}{\partial n} = \mathbf{n} \cdot \left[ -\sum_{q=0}^{J_e-1} \beta_q \frac{\partial \mathbf{u}^{n-q}}{\partial t} - \sum_{q=0}^{J_e-1} \beta_q N(\mathbf{u}^{n-q}, \mathbf{w}^{n-q}) \right]
\]

\[
- \mathbf{n} \cdot \left[ \frac{1}{Re} \sum_{q=0}^{J_e-1} \beta_q [\nabla \times (\nabla \times \mathbf{u}^{n-q})] \right],
\]

\[
\nabla^2 p^{n+1} = \nabla \cdot \left( \frac{\mathbf{u}^{n-q}}{\Delta t} \right),
\]

\[
\nabla^2 \mathbf{u}^{n+1} - \frac{\gamma_0 Re}{\Delta t} \mathbf{u}^{n+1} = -\frac{Re}{\Delta t} (\mathbf{u} - \Delta t \nabla p^{n+1}),
\]

\[
\nabla^2 \mathbf{w}^{n+1} = 0,
\]

where \(N\) denotes the nonlinear (inertial) term \((\mathbf{u} - \mathbf{w}) \cdot \nabla \mathbf{u}\), and \(\mathbf{n}\) is the outward unit normal on the domain boundaries and \(\Delta t\) is the time-step. The first three steps are explicit and computed using the values of \(\mathbf{u}\) and \(\mathbf{w}\), which are computed at the quadrature points. The last three steps are computed in a variational framework. Details of the methods used to solve the matrix systems resulting from the corresponding variational statements can be found in Ref. 20. The constants \(\alpha_q, \beta_q\) are integration weights and are defined in Ref. 19. The mesh velocity is in general arbitrary, and can be specified explicitly or can be obtained from a Laplace equation.

5. Numerical Simulation Results

The incompressible Navier–Stokes equations in the micro-pump system are solved by the \(\mathcal{N}_{\varepsilon\kappa\mathcal{T}or–ALE}\) model described in the previous section. The algorithm
is shown to maintain the numerical accuracy and convergence under relatively large mesh stretching conditions, avoiding expensive remeshing procedures for most applications. In our simulations, full closure of the inlet and exit valves require *annihilation* of the elements trapped between the valve tips and the top wall, requiring remeshing of the computational domain. Here we avoided remeshing by allowing a gap between the valve tips and the top wall. This gap was maintained at $g_{\text{min}} = 0.025L$ for closed valve positions.

The spectral element mesh used in this study is presented in Fig. 3 at various instants, corresponding to the suction and ejection stages. The flow domain is represented with 222 unstructured spectral elements. Each element utilized 7th-order polynomial modal expansions. Accuracy of our solution is checked by successive p-refinements (by increasing the polynomial order per element). For spectral element formulations, the discretization error decreases exponentially fast with p-refinements. In fact, we have not observed any changes between the numerical results of the 5th-order and 7th-order expansions. The typical discretization errors are well below 1%.

In Fig. 4, we present instantaneous volumetric flowrate for two different Reynolds numbers ($Re = 3$ and 30). The volumetric flowrate of fluid entering through the inlet valve is shown as positive and the flow leaving is indicated as negative. The sum of the two is the rate of change of the control volume due to the oscillation of the membrane,

$$\frac{dV}{dt} = \dot{Q}_{\text{in}} - \dot{Q}_{\text{out}}.$$
The membrane motion is periodic. Therefore, the net amount of fluid displaced by the membrane during a period is zero

\[ \int dV = 0 = \int_0^\tau \dot{Q}_{\text{in}} \, dt - \int_0^\tau \dot{Q}_{\text{out}} \, dt. \]

In other words, \( \dot{Q}_{\text{in}} = \dot{Q}_{\text{out}} = \dot{Q} \). Numerically integrating the curves under inlet and exit valves in Fig. 4, we determined the effective flowrate in our micro-pump. The ratio of the numerically calculated flowrate to the maximum flowrate given by Eq. (7) defines the efficiency (\( \eta \)) of the micro-pump. The results are presented in Table 1. It is clearly seen that the efficiency of the pump decreases with increasing Reynolds number (\( Re = \omega a^2 / \nu \)). This efficiency decrease is due to the imperfect closure of the valves, where there is a finite gap between the top wall and the valves at valve closed position, hence the fluid leaks. The average flowrate of the pump increases with the Reynolds number, as predicted by Eq. (7). This is either due to increase in the size of the pump (increase in \( L \)) or increase in the frequency \( \omega \) for a given fluid (fixed \( \nu \)). For fabrication of our conceptual design, the actual dimensions of the micro-pump can be determined by either selecting the membrane length \( L \) or the actuation frequency \( \omega \). For example, choosing water and selecting...
Table 1. The actuation frequency, predicted and computed mass flowrate (per unit width) and the efficiency of the micro-pump as a function of the Reynolds number. The data is obtained for water and $L = 100 \mu m$.

<table>
<thead>
<tr>
<th>$Re$</th>
<th>$\omega$ [kHz]</th>
<th>$\dot{M}/W_{Th}$</th>
<th>$\dot{M}/W_{Num}$</th>
<th>$\eta$</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.3</td>
<td>3</td>
<td>0.0382</td>
<td>0.0351</td>
<td>92%</td>
</tr>
<tr>
<td>3</td>
<td>30</td>
<td>0.382</td>
<td>0.336</td>
<td>88%</td>
</tr>
<tr>
<td>30</td>
<td>300</td>
<td>3.82</td>
<td>2.86</td>
<td>75%</td>
</tr>
</tbody>
</table>

the membrane length to be 100 $\mu m$ we calculated the frequency of operation and the corresponding mass flowrate in the pump. The results are also presented in Table 1. The power required to drive the pump can easily be obtained by integrating the product of the numerically obtained pressure and the velocity on the membrane.

Parametric studies of the normal and tangential fluid forces on the deflecting membrane are performed as a function of the Reynolds number ($Re$). For viscous flow, total normal stress acting on the membrane has two components; pressure and the viscous normal stresses. The viscous normal stress acting on the membrane is about three orders of magnitude less than the pressure, as computed in our simulations. Therefore, we mainly focus on the pressure acting on the membrane (the normal force per unit area), and the shear stress (viscous tangential force per unit area).

In Fig. 5, we present the pressure distribution on the membrane as a function of time ($\tau\omega$) for two different Reynolds number simulations. The position of the membrane during the periodic motion was presented in Fig. 2. The pressure is normalized with the dynamic head $P^* = P/(2\rho(\omega\tau)^2)$. For $Re = 3$, pressure acting on the membrane is more uniformly distributed at a given time than the $Re = 30$ simulation. In fact, the pressure distribution shows strong spatial variations for $Re = 30$, where the inertial effects dominate the flow.

In Fig. 2, we observe that the membrane reaches its maximum deformed position at $\tau\omega = 0.25$ and $\tau\omega = 0.75$. The position, velocity and the acceleration of the membrane was given by Eq. (1). For fluid structure interaction, we expect the maximum normal force acting on the membrane correspond to the instant when the membrane acceleration is maximum. However, we observed from Fig. 5 that the maximum and minimum pressure occur at $0.35\tau\omega$ and $0.85\tau\omega$, respectively. This is an interesting result. We investigated the reasons of the phase shift of $0.1\tau\omega$, and have determined that the inlet and the exit valves affect the pressure build-up in the system. If we examine the inlet and exit valve motion (see Fig. 2, bottom), we
Fig. 5. Nondimensional gauge pressure variation on the membrane as a function of the period $(\tau \omega)$.

(a) $Re = 3.0$.

(b) $Re = 30.0$. 
observe that the inlet valve fully opens (and the exit valve fully closes) at $0.35\tau\omega$. This is the beginning of the effective suction stage. During $0.20 \leq \tau\omega \leq 0.35$ the mean flow direction changes from ejection to suction direction, combined with the changes due to the variation of the inlet- and exit-valve gaps. The membrane overcomes the inertia of the fluid during this transient motion, resulting in a pressure build-up at $\tau\omega \approx 0.35$. A similar process happens at $\tau\omega \approx 0.85$ (beginning of the ejection stage). We have verified this claim by opening the inlet and exit valves by an equal amount, and actuating the membrane. While the valves were not moving, we obtained maximum pressure build-up at $\tau\omega = 0.25$, and minimum pressure at $\tau\omega = 0.75$, as expected.

The shear stress variation on the membrane, normalized with the dynamic head $\tau^* = \tau/(2\rho(\omega_0)^2)$, is presented in Fig. 6. The shear stress is about two orders of magnitude less than the pressure. Since the shear stress acts in the tangential direction to the membrane, it will change the local tension on the membrane. This may have important implications in coupled fluid/structure interaction problems. Here, we imposed the motion of the membrane as a function of $\omega = \pi c/L$, where $c^2 = T/M$ and $T$ is tension of the membrane that is assumed to be a constant.

A coupled fluid-structure interaction study will reveal some secondary mode of vibration and shearing/distortion of the membrane. Especially, this will become more important at high Reynolds number applications (noting that the maximum shear stress is about $1/40$ and $1/100$ of the maximum pressure for the $Re = 30$ and $Re = 3$ cases, respectively). For relatively small Reynolds number cases used in this study, this effect is negligible.

The vorticity contours are presented in Fig. 7 for $Re = 30$ flow at various times. Figure 7(a) is at $\tau\omega = 0.28$ and it corresponds to the beginning of the suction stage. Start-up vortices due to the opening of the inlet valve can be identified as a vortex pair just at the top of the inlet valve. Figure 7(b) corresponds to the end of the suction stage (at $\tau\omega = 0.72$). A vortex pair is visible in the pump cavity. The flow pattern at $\tau\omega = 0.84$, corresponding to early ejection stage is given in Fig. 7(c). The exit valve has just opened, and the start-up vortex due to its motion is visible at the top of the exit valve. Meanwhile, the vortex pair in the pump cavity has evolved further. The negative vortex is trapped in the pump cavity, and the positive vortex is deflected from the membrane towards the middle of the pump cavity. Presence of the vortex pair also creates strong vorticity on the membrane wall. The vorticity $\Omega$ on the membrane is proportional to the shear stress $\tau_W$ on the membrane ($\Omega = -\tau_W/\mu$). Strong shear stress variations on the membrane, presented in Fig. 6 at $\tau\omega \approx 0.85$, indicates this situation. Throughout the entire period of the micro-pump operation, the negative vortex is present in the pump cavity close to the inlet valve (as seen from Fig. 7). Its strength varies during the cycle. However, it is continuously re-enforced at the suction stage. Although, the presence of this vortical structure adversely affects the performance of the micro-pump, it is a desired feature if the micro-fluidic system is to be utilized as a micro-mixer.
Fig. 6. Nondimensional shear stress variation on the membrane as a function of the period ($\tau \omega$).

(a) $Re = 3.0$.

(b) $Re = 30.0$. 

Nondimensional shear stress variation on the membrane as a function of the period ($\tau \omega$).
After demonstrating the performance of the micro-pump with two-dimensional simulations, we also tested the design with three-dimensional simulations. We extended the two-dimensional geometry of the micro-pump uniformly, using $W = 0.4L$. We observed boundary layer growth on the side walls, which reduces the flowrate slightly. Nevertheless, the three-dimensional performance of the micro-pump is acceptable.
Finally, the algorithm is fast enough to test the two-dimensional conceptual design on a work station. Low Reynolds number \((Re \leq 5)\) simulations require about 222 triangular spectral elements with 5th order expansions. The simulation takes about 0.6 CPU seconds per time step in a 195 MHz Silicon Graphics Onyx2 work station. The entire simulation for \(Re \leq 5\) requires 2 000 time steps per period of micro-pump operation, resulting in approximately 20 min per run. However, three-dimensional simulations require large memory and CPU times. A parallel version of the algorithm is used to test three-dimensional effects on several design cases. For three-dimensional runs, we have used 444 prism-type spectral elements, utilizing 5th order polynomial expansion per element, resulting in 90 calculation points per element. The computational domain is divided into 8 sub-domains and the parallel simulations are performed at IBM SP2 (thin nodes) at Brown University, Center for Fluid Mechanics. We observed 18 CPU seconds per time-step for the parallel runs. Good parallel efficiencies are obtained.

6. Discussions

We presented the conceptual design and numerical simulation-based verification of a bidirectional micro-pump. The flow direction can be reversed simply by reversing the motion of the valves. Reversibility of the pump is an advantage compared to the conventional hanging-beam type micro-valves. Hence, our micro-pump can also be utilized as a micro-mixer. For high Reynolds number applications, presence of vorticies in the pump chamber can significantly enhance micro-scale mixing. The piston type inlet and exit valves used in our design could be fabricated as a free floating gate-valve, where thermally generated vapor bubbles enable the valve motion, as recently demonstrated by Papavasiliou et al. 21

Over all, the micro-pump design requires utilization of micro-scale hydraulic systems for force and motion transfer. Numerical simulations based on the ALE schemes can help analyze the motion of micro-hydraulic piston arrangements. Viscous loads and piston leakage problems can be easily analyzed by our current formulation. If micro-hydraulics can deliver similar advantages of the macro scale hydraulic devices, a new means of micro-actuation can be implemented in emerging MEMS and micro-fluidic technologies.

Finally, we demonstrated that CFD simulations can be utilized to validate various micro-fluidic concepts on a computer, prior to hardware fabrication and experimental verification. Since two-dimensional simulations of a micro-fluidic concept on a work station typically takes about half an hour, CFD simulations can be effectively utilized to save time and resources during micro-fluidic design.

Acknowledgments

We would like thank Prof. George Em Karniadakis of Brown University for allowing us to use his computational facilities at Center for Fluid Mechanics. T. C. W. acknowledges DARPA grant 49620-96-1-0 and DOE grant DE-FG02-98ER253. A. B. acknowledges DELL STAR program for their support during the course of this work.
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