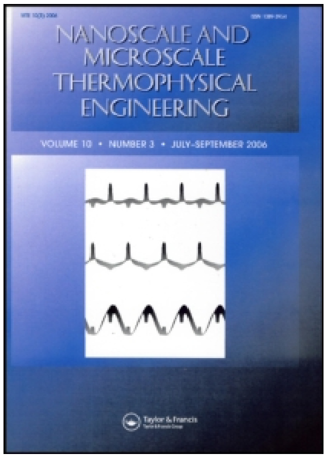

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Zhi-Xin Li [a]
[a] Department of Engineering Mechanics, Tsinghua University, Beijing, P. R. China.

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PUMPING MECHANISM OF THERMALLY DRIVEN PHASE TRANSITION MICROPUMP

Zhi-Xin Li, Mo-Ran Wang, Xiao-Bo Yao, and Zeng-Yuan Guo
Department of Engineering Mechanics, Tsinghua University, Beijing, P. R. China

The theoretical analysis and the experimental investigation for the thermally driven phase transition micropump are performed. Firstly, single-phase flow driven by a moving heat source is simulated to reveal the pumping effect. Then, a simplified model for the thermally driven phase transition micropump is established in a coordinate system moving with the heater. The pumping mechanism and characteristics are analytically and numerically analyzed. The results indicate that the thermally driven flow comes from the asymmetrical fluid property distributions, and an optimal distribution of phases is essential to get the best pumping effect. Experimental results show that an optimum heating and cooling condition is necessary for the maximum flow rate, which proves the rationality of the analysis.

Keywords pumping mechanism, thermally driven, micropump, phase transition

1. INTRODUCTION

Micropumps are important actuators used in the micro-electro-mechanical systems (MEMS), and the relevant technology is critical for the micro fluidic systems development. Micropumps are divided into two types, depending on whether or not they have valves. The valveless micropumps have attracted much attention, due to the good performance and suitability for micromation. Up to now, numbers of new micropumps, such as thermopneumatic micropump [1], electrohydrodynamic (EHD) micropump [2], magneto-hydrodynamic (MHD) micropump [3], etc. have been developed.

The thermally driven phase transition micropump is a novel valveless micropump, in which fluid in microchannels can quickly evaporate or condense, due to the smaller thermal inertia. Yuan et al. [4] reported that when the growth and collapse of a single bubble did not occur at the tube midpoint, a net pumping effect was then encountered. Tsai et al. [5] utilized the bubble expansion and shrinkage to drive a nozzle-diffuser micropump. Mizoguchi [6] built the first light driven micropump, in which laser was used to make liquid change phase to push the fluid to higher pressure. Figure 1 shows

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Address correspondence to Zhi-Xin Li, Department of Engineering Mechanics, Tsinghua University, Beijing, 100084, P. R. China. E-mail: lizhx@tsinghua.edu.cn

NOMENCLATURE

A	cross-sectional area of tube [m^2]	r	latent heat [kJ/kg]
C_v	specific heat at constant volume [J/(kg·K)]	R	molar gas constant [J/mol·K]
D	microtube diameter [m]	S_h	heat source intensity [W/m^3]
f_u	resistance coefficient from radial viscous diffusion	T_a	surrounding temperature [K]
		T_s	temperature of two-phases section [K]
h	heat transfer coefficient of the outer tube surface [W/m^2·K]	U	fluid velocity relative to the heater [m/s]
		U_h	velocity of the moving heater [m/s]
h_P	$h_P = h\frac{P}{A}$, [W/m^3·K]	ϕ	quality
k	thermal conductivity [W/m·K]	Φ	viscous dissipation per volume [W/m^3]
L	distance between heaters; tube length [m]	μ	dynamic viscosity [kg/(m·s)]
L_l	length of the liquid section [m]	μ_m	average dynamic viscosity [kg/(m·s)]
L_t	length of the two-phase section [m]	ν_m	average kinetic viscosity [m^2/s]
L_g	length of the gas section [m]	ρ	density [kg/m^3]
m	mass flux [kg/m^{2}s]	ρ_m	average density [kg/m^3]
P	microtube perimeter [m]	η_f	mass flow coefficient
p_0	pressure at S.T.P. [1 atm]		

the structure of the phase transition micropump, which consists of a microtube and an array of heating elements. When the heaters are powered in a cyclically scanning manner, as indicated in Figure 1, phase transition occurs in the heating section. The thermophysical properties are different at the two ends of the two-phase column, due to the heater movement, which causes a net flow towards the scanning direction. Ozaki et al. [7–9] did pioneer work on the phase transition micropump. They assumed that three fluid parts (liquid, gas, liquid) flowing in the tube were time-independent, and the two-phase region was ignored. They ascribed the pumping effect to the large kinematic viscosity difference between liquid and gas. Wang and Li [10] presented a simple model

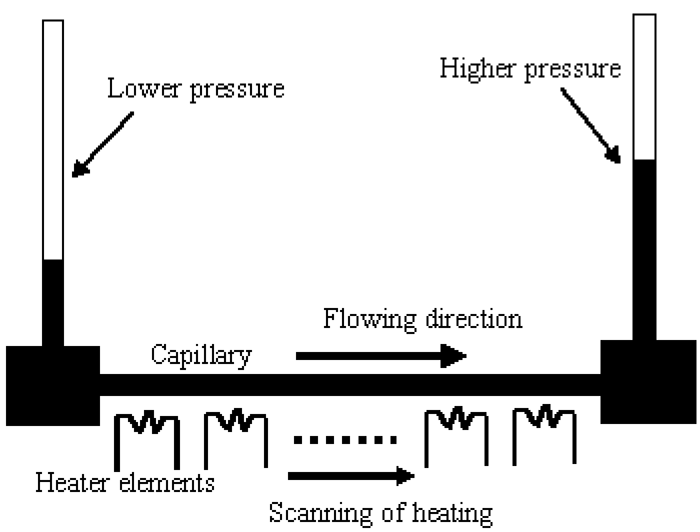

Figure 1. Schematic of the phase transition micropump.

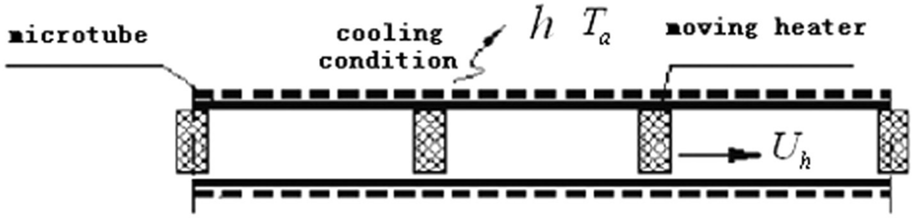

Figure 2. Single phase tube flow with moving heat sources.

for the traveling phase transition in micro tubes and concluded that the instantaneous high pressure during evaporation is the main force driving the pump. Based on the theoretical and experimental investigations, this study presents a detailed analysis on the pumping mechanism of the phase transition micropump.

2. SINGLE PHASE FLOW

Single-phase flow in a tube with moving heat sources was firstly investigated to reveal the pumping effect of moving heating sources. The physical model is simplified as a long tube with periodical moving heaters, as shown in Figure 2. The heaters move at a speed of U_h. The tube is cooled under natural convection with a heat transfer coefficient per length, h_l and a surrounding temperature T_a.

The governing equations of the single-phase flow are as follows:

$$\rho_t + (\rho u)_x = 0 \tag{1}$$

$$(\rho u)_t + (\rho u^2)_x = -f_u u + \mu u_{xx} - p_x \tag{2}$$

$$\rho C_v T_t + \rho u C_v T_x + p u_x = k T_{xx} + h_P (T_a - T) + S_h + \Phi \tag{3}$$

$$p = \rho R T \tag{4}$$

where the subscripts x and t represent partial derivatives, and $h_p = hP/A$.

In order to simplify the analysis, a coordinate system moving with the heat source is established by the transformation expressions:

$$U = u - U_h \tag{5}$$

$$X = x - U_h t \tag{6}$$

Ignoring the time-dependent terms, the governing equations in the moving coordinate system become:

$$\rho U = m \tag{7}$$

$$\int_0^L U dX = -U_h L \tag{8}$$

$$\rho U C_v T_X + p_0 U_X = k T_{XX} + h_P (T_a - T) + S_h + \Phi. \tag{9}$$

The pressure in the whole tube is simplified uniform:

$$p_0 = \rho R T \tag{10}$$

Combination of Eqs. (5)–(10) obtains the ordinary differential equation of $\frac{1}{\rho}$:

$$\left(\frac{1}{\rho}\right)_{XX} \cdot \frac{k}{R} - \left(\frac{1}{\rho}\right)_{X} \cdot m\frac{C_p}{R} - \left(\frac{1}{\rho}\right) \cdot \frac{h_P}{R} + \frac{h_P T_a + S_h + \Phi}{p_0} = 0 \tag{11}$$

The gas velocity is:

$$u = \frac{m}{\rho} + U_h \tag{12}$$

Equations (11) and (12) are numerically calculated with a periodic boundary condition, which is:

$$\rho(0, t) = \rho(L, t) \tag{13}$$

The parameters used in the calculations are $D = 1$ mm, $L = 10$ mm, $U_h = 1$ m/s, $T_a = 300$ K, and $p_p = 10^5$. The S_h is simplified as 10^7 W/m^3, and h equals 20 W/m^2·K. The properties of air were consulted in Yang's Book [11]. Figure 3 shows the results for the density and the velocity versus time at $x = 0$ in a heating cycle. The gas density and velocity are asymmetrical in a cycle, due to the moving heater. The mass flow towards the positive x direction is greater than the mass flow towards the negative x direction. As a result, a net mass flow is obtained in the heat source moving direction.

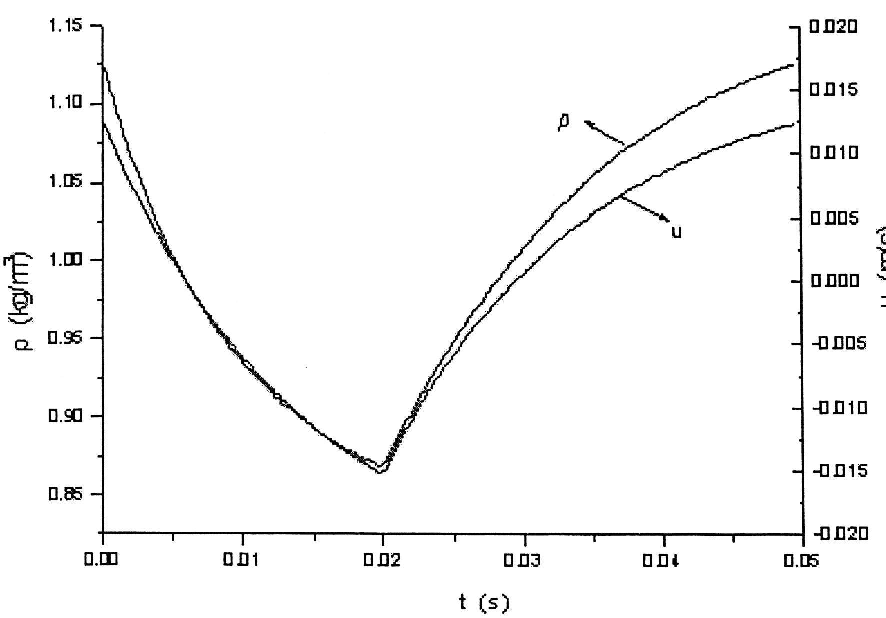

Figure 3. Variation of gas density and velocity in a cycle at $x = 0$.

Nevertheless, the net mass flow rate is too small, even though the heat source is very large for single-phase gas flow.

3. MECHANISM ANALYSIS FOR PHASE TRANSITION MICROPUMP

The flow in the phase transition micropump is very complicated. It is an unsteady process, which includes single-phase flow, two-phase flow, and evaporation and condensation. Like the analysis for the single-phase flow, a moving coordinate system is helpful. In the moving coordinate system, the heat sources are fixed, while the tube moves at a speed of U_h towards the negative x direction, as shown in Figure 4. Fluid moves at a speed of U relative to the heat source. Liquid vaporizes near the heat source and condenses before the next heat source coming, which means that the whole tube contains gas, two-phase, and liquid sections, as illustrated in Figure 4. L_l, L_t, and L_g are lengths of the liquid, two-phase, and gas sections, respectively. The heat transfer coefficient of the outer tube surface is h, and the surrounding temperature is T_a. Because the heat source is fixed at $x = 0$ in the moving coordinate system, the periodical condition between 0 and 1 is adopted. That is, the fluid density and viscosity are the same at $x = 0$ and $x = 1$. The one-dimensional governing equations are:

Continuity equation;

$$\rho U = \text{const} \tag{14}$$

Momentum equation;

$$\Delta p = \int_0^L \frac{32\mu}{D^2}(U + U_h)\, dX \tag{15}$$

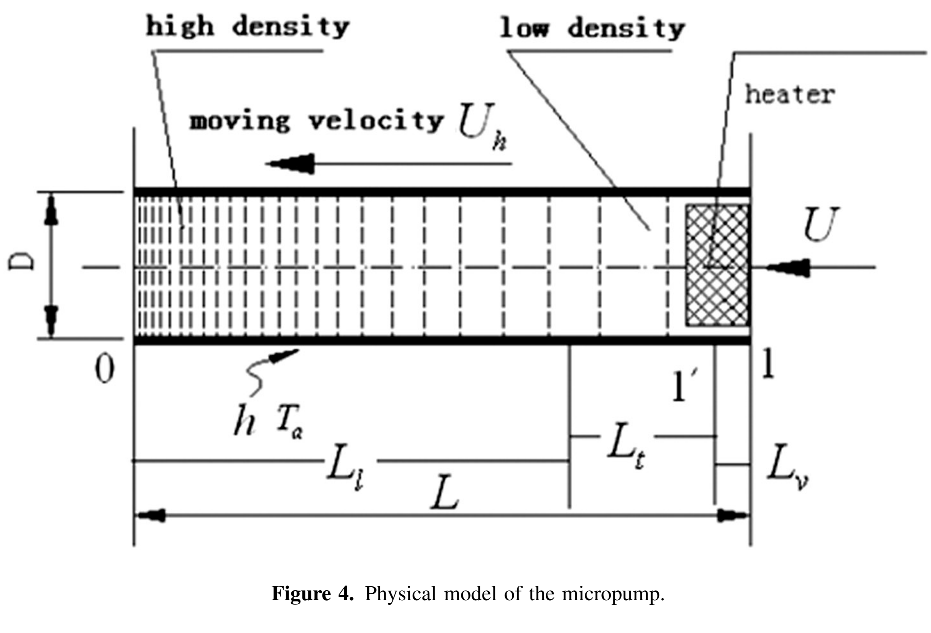

Figure 4. Physical model of the micropump.

where the inertial item integral $\rho U \frac{dU}{dX}$ from 0 to 1 does not appear, because it equals to zero under a periodical boundary assumption. In general, Eq. (15) is valid only for the single phase regions in a microtube. Nevertheless, Eq. (15) can be regarded to be valid for the whole tube, due to the length of the two-phase region, which is relatively smaller to that of the single phase region. However, Eq. (15) is also assumed valid for two phase flow in the microtube. By using the subscript m to express the value averaged over the microtube length, there are:

$$\rho_m = \left(\int_0^L \rho dX \right) \Big/ L \tag{16}$$

$$\mu_m = \left(\int_0^L \mu dX \right) \Big/ L \tag{17}$$

$$\nu_m = \left[\int_0^L (\mu/\rho) dX \right] \Big/ L \tag{18}$$

Combining Eqs. (14)–(18), we can express the mass flow as:

$$m = \left(\rho_m - \frac{\mu_m}{\nu_m} \right) U_h + \frac{\Delta p D^2}{32 L \nu_m} \tag{19}$$

where D is the diameter of the microtube; Δp is the pressure drop between the inlet and outlet.

Equation (19) indicates that the mass flow in the microtube is composed of two terms: one is driven by the pressure drop, Δp; the other, $(\rho_m - \mu_m/\nu_m)U_h$, is driven by the moving heat source. Once the phase transition occurs by moving heat sources and essential cooling conditions, the fluid properties rapidly change before and behind the moving heat sources. The fluid property asymmetry leads to a unidirectional flow. When thermally driven flow is more than inverted pressure driven flow (in Eq. 19, $\Delta p < 0$), a pumping effect is encountered.

4. EFFECTS OF LIQUID, TWO-PHASE AND GAS SECTIONS ON THE PUMPING FLOW

In fact, the flow rate of the phase transition micropump depends not only on the moving heat source, but also on the cooling conditions. However, there is no obvious mass flow, if the heat source intensity is too small to evaporate the liquid, or if the heat source is too strong, relative to the cooling conditions. As a result, the pump does not work.

Consider a linear correlation between the quality and the spatial coordinate one obtains:

$$\Delta \phi = a L_t. \tag{20}$$

By using the Medam equation:

$$\frac{1}{\rho_{1,2}} = \frac{\phi}{\rho_2} + \frac{1-\phi}{\rho_1} \tag{21}$$

$$\frac{1}{\mu_{1,2}} = \frac{\phi}{\mu_2} + \frac{1-\phi}{\mu_1} \tag{22}$$

where the subscript 1 represents gas phase properties and the subscript 2 represents liquid phase properties, Eqs. (16)–(18) become:

$$\rho_m = \rho' \cdot \frac{L_l}{L} + \frac{\rho'\rho''}{\rho' - \rho''} \ln\left(1 - \Delta\phi + \frac{\rho'}{\rho''}\Delta\phi\right) \cdot \frac{L_w}{L} + \rho'' \cdot \frac{L_v}{L} \tag{23}$$

$$\mu_m = \mu' \cdot \frac{L_l}{L} + \frac{\mu'\mu''}{\mu' - \mu''} \ln\left(1 - \Delta\phi + \frac{\mu'}{\mu''}\Delta\phi\right) \cdot \frac{L_w}{L} + \mu''\frac{L_v}{L} \tag{24}$$

$$\begin{aligned} \nu_m = {} & \frac{\mu'}{\rho'} \cdot \frac{L_l}{L} + \frac{\mu''}{\rho''} \cdot \frac{L_v}{L} + \frac{(\rho' - \rho'')\mu'\mu''}{\rho'\rho''(\mu' - \mu'')} \cdot \frac{L_t}{L}\Delta\phi \\ & + \frac{(\rho''\mu' - \rho'\mu'')\mu'\mu''}{\rho'\rho''(\mu' - \mu'')^2} \ln\left(1 - \Delta\phi + \frac{\mu'}{\mu''}\Delta\phi\right) \cdot \frac{L_w}{L} \end{aligned} \tag{25}$$

where L_w is defined as $\frac{L_t}{\Delta\phi}$. The superscript $'$ represents liquid phase, and $''$ represents gas phase.

The numerical results of $\rho_m - \mu_m/\nu_m$ versus L_l/L_t for different L_v/L_t are shown in Figure 5. It shows that when the two-phase section exists ($L_t \neq 0$), $\rho_m - \mu_m/\nu_m$ is always positive. That is, when the moving heat source is strong enough to vaporize the liquid, and the vapor can be condensed under the cooling condition, the fluid will flow towards the heater moving direction under zero pressure drop. The mass flow rate is dependent on the distribution of the lengths of the liquid, two-phase, and gas sections. There exists an optimal distribution for the three section lengths, where the flow rate reaches a maximum. Define:

$$m_0 = \frac{4h(T_s - T_a)L}{Dr} \tag{26}$$

as the characteristic flow rate, which represents the mass of the saturated vapor condensed to saturated liquid under a proper cooling condition. Consequently, the mass flow rate of the micropump can be expressed as:

$$m = \frac{\Delta p D^2 \rho_m}{32L\mu_m} + m_0 \cdot \frac{L_t}{L\Delta\phi}\left(\frac{\rho_m \nu_m}{\mu_m} - 1\right) \tag{27}$$

where h is the heat transfer coefficient, and r is the latent heat of the working fluid. $\Delta\phi$ is the quality variation in the tube. Define the flow rate coefficients as:

$$\eta_f = \frac{L_t}{L\Delta\phi}\left(\frac{\rho_m \nu_m}{\mu_m} - 1\right). \tag{28}$$

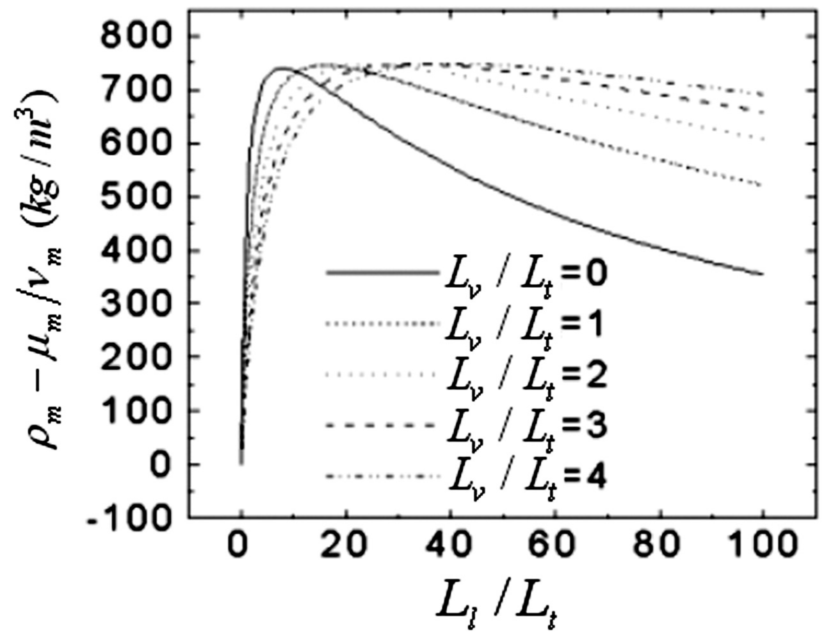

Figure 5. Variation of $\rho_m - \mu_m/v_m$ with the ratio of liquid to two-phase lengths.

The maximum pumping flow rate in the case of $\Delta p = 0$ can be expressed as:

$$m_{\max} = m_0 \eta_f \tag{29}$$

Figure 6 shows the variations of flow rate coefficients η_f with L_l/L_t for different L_v/L_t. An optimal L_l/L_t exists for the maximum flow rate. In fact, the optimal L_l/L_t requires that the heating power should be matched with the cooling conditions. If the heating power is too large, the whole microtube will be filled with vapor. However, if the heating power is very small, the tube will be entirely full of water. No thermally pumping flow can be obtained in both cases.

5. EXPERIMENT

The experiments for the micropump were done to confirm the theoretical analysis. Figure 7 shows the experimental system. The stainless steel microtube, used as the body of micropump, is 200 μm in inner diameter and is divided into 5 parts for electrical heating. The working fluid is the distilled water. By scanning electric current supplied cyclically through the stainless steel microtube, the fluid in the tube can be evaporated/condensed and pumped to the scanning direction. The measured pumping flow rates under different heating and cooling conditions are shown in Figure 8, where the switch time is the duration for heating one subsection in a cycle. The results show that there is an optimal heating current to gain the maximum flow rate for a given cooling condition. This is in good agreement with the analytical results.

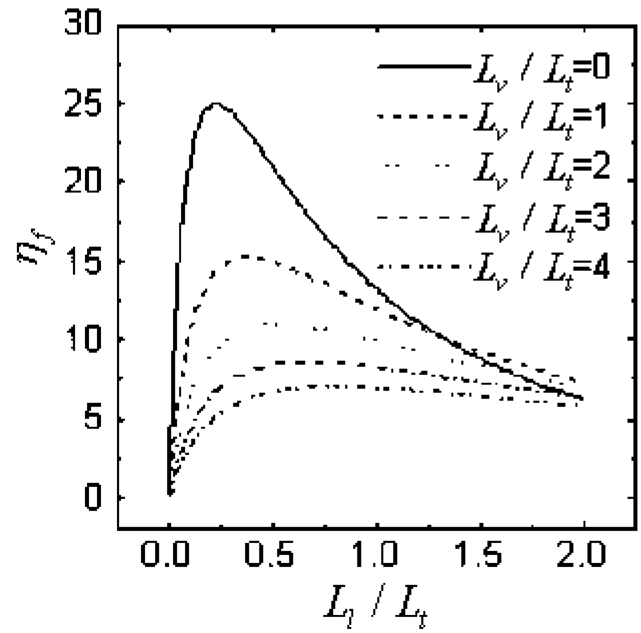

Figure 6. Variation of flow rate coefficient with the ratio of liquid to two-phase lengths.

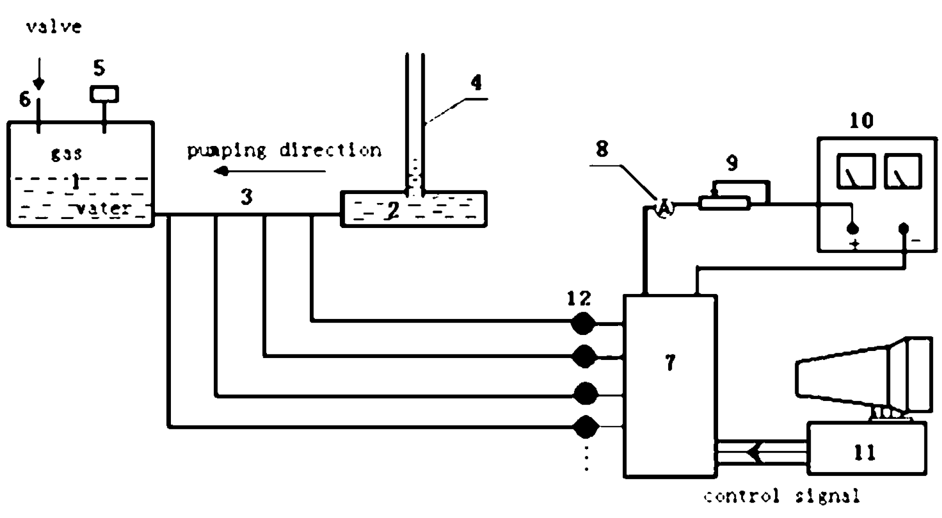

Figure 7. Experimental system.

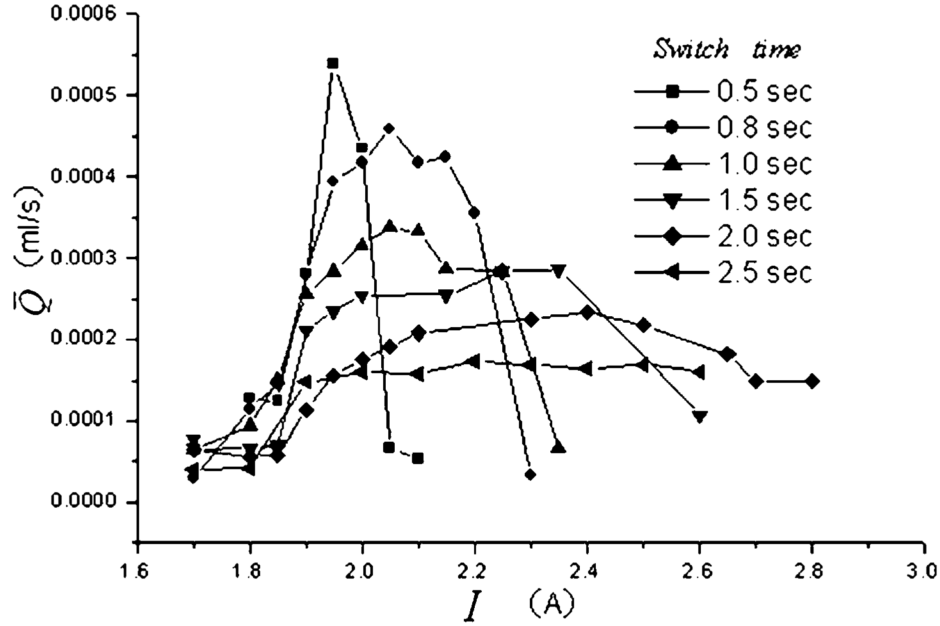

Figure 8. Mass flow versus the heating currents.

6. CONCLUSIONS

Theoretical analysis and experimental investigation for the thermally driven phase transition micropump are performed. The thermally driven flow is dependent on the velocity of moving heat sources and the asymmetrical distribution of fluid properties. Once evaporation and condensation occur by moving heat sources and cooling conditions, the fluid properties are no longer symmetrical before and behind the moving heaters. When the thermally driven flow is more than the inverted pressure driven flow, a net unidirectional pumping flow is obtained. A reasonable coordination of the heating condition, cooling condition, and the heater moving velocity, which result in an optimal ratio of two-phase to liquid lengths, will lead to the maximum flow rate. The experimental results show that there is an optimum heating current to get the maximum flow rate, which is in good agreement with the analytical results.

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