A valve-less planar fluid pump with two pump chambers

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Abstract

A new planar fluid pump based on the valve-less diffuser/nozzle pump principle is presented. The pump consists of two pump chambers, each with two flow rectifying diffuser/nozzle elements with rectangular cross sections, one at the inlet and one at the outlet. The pump chambers are arranged in parallel for high pump flow. Each pump chamber has two piezoelectrically vibrated diaphragms. The planar pump is fabricated in brass with a total thickness of 1 mm. The pump chamber diameter is 13 mm and the diffuser/nozzle element neck dimensions are 0.3 x 0.3 mm. Simplified theoretical analyses of the maximum pump flow and resonance frequency are given. The flow rectifying ability of the diffuser/nozzle elements is demonstrated in a stationary flow situation and the pump performance is verified in two different pump mode configurations: anti-phase and in-phase chamber volume excitation. The measurements in the anti-phase mode show pump flows and pump pressures which are more than twice as high as those of the in-phase oscillation mode. The anti-phase mode has a pump capacity of about 16 ml/min and a maximum pump pressure of about 1.7 m H2O with the pump diaphragm vibration frequency set to the pump resonance frequency of 540 Hz.

Keywords: Planar fluid pump; Pumps

1. Introduction

There is great and increasing interest in making smaller fluid pumps [1]. Many of these so-called micropumps are designed to handle small and precise volumes for different chemical, medical and biomedical applications. Most of these pumps are diaphragm pumps with two passive check valves [2-10]. However, pumps with movable parts, such as check valves, have many drawbacks. There is a risk of reduction in performance and reliability due to wear and fatigue. High fluid pressure loss at the check valves may also be a problem for pumps using movable parts. There is also a considerable risk of valve clogging.

We have recently presented a novel nozzle/diffuser-pump principle with no moving parts and with very good pump performance in terms of pump flow and pressure [11,12]. Instead of passive check valves the pump uses two specially designed nozzle/diffuser elements which give a fluid directing effect as shown in Fig. 1.

One of the main advantages of the new pump principle is that the pumps can be fabricated as micropumps. Here we describe the design and results of an improved version of the pump based on the diffuser/nozzle principle. The main features of the new pump are a flat, planar design only 1 mm thick. The pump is arranged in a parallel configuration of two identical pump units where each unit consists of a pump chamber with two diffuser/nozzle elements, one at the inlet and one at the outlet.

2. The pump principle

The basic elements in the diffuser/nozzle pump unit consist of two diffuser/nozzle elements connected to a fluid cavity volume with an oscillating diaphragm as shown schematically in Fig. 1. A diffuser is defined as a conduit with an expanding cross-sectional area in the flow direction and a nozzle is a conduit with a decreasing cross-sectional area in the flow direction. The pump operation is based on the fluid flow rectifying properties
of the two nozzle/diffuser elements. Providing that the diffuser/nozzle element is correctly designed, the volume flow in the diffuser direction is higher than the volume flow in the nozzle direction, assuming the same pressure drop across the fluid element. The pump cycle of the pump can be divided into a 'supply mode' and a 'pump mode'. During the 'supply mode' the cavity volume increases and a larger amount of fluid flows into the cavity through the input element, which acts as a diffuser, than through the output element, which acts as nozzle. However, during the 'pump mode', when the cavity volume decreases, a larger amount of fluid flows out of cavity through the output element, which acts as a diffuser, than through the input element, which acts as a nozzle. The result for the complete pump cycle of the diffuser/nozzle pump is that a net volume has been transported from the input to the output side of the pump.

This valve-less pump has a pump resonance frequency which is mainly controlled by the mass of the fluid in the diffuser/nozzle element and the elastic properties of the diaphragm. An increased pump frequency means an increased pump flow. Best pump performance is achieved by having an excitation frequency which is close to the resonance frequency. The resonance frequency is typically several hundreds of Hertz, which is more than one order higher than the optimum drive frequencies of check valve pumps of comparable size [2–10]. The possibility to have a high excitation frequency is the main reason why this pump principle has shown much higher pump flow performance than micropumps based on passive check valves.

3. The planar two-chamber pump

In order to improve further the pump performance and to facilitate miniaturization we have designed a planar pump with two pump chambers, shown in Fig. 2. There are two principal types of diffuser geometries: the conical and the rectangular [13]. We used the conical diffuser design in the first presented single-chamber diffuser/nozzle pump. In this new planar version we employ the rectangular diffuser design with four flat walls; two parallel and the other two diverging.

The pump was fabricated from 0.5 mm thick plates of brass into which the pump chamber cavities and diffusers were milled in one milling sequence to a common depth of 0.15 mm. A top view of the plate geometry with the chambers and diffusers is shown in Fig. 2(a). The pump chamber diameter is 13 mm and the diffuser inlets are slightly rounded with a neck width of 0.3 mm and a diffuser outlet width of 1.0 mm. The diffuser length is 4.1 mm. In each plate, two pump chamber cavities, one inlet and one outlet cavity, were formed. Since the bottoms of the pump cavities also constitute the diaphragms, the resulting diaphragm thickness was 0.35 mm. One inlet and one outlet flow hole were drilled in the inlet/outlet cavity of each half before the two plate halves were fixed together using epoxy. Two short brass pipes were fixed to the pump inlet and outlet holes for the tube attachment. Four PZT-piezoelectric discs, 10 mm in diameter and 0.2 mm thick, were fixed to the four pump diaphragms by epoxy. This resulted in an electrical contact between the metal plate structure and one of two electrodes of each piezoelectric disc.

The two planar pump units of this pump were arranged in a configuration corresponding to a parallel connection of two single-chamber pumps, as illustrated in Fig. 3. This would in principle ensure a pump volume flow of twice that of a single-chamber pump with the same excitation and dimensions. However, in the single-chamber pump the diaphragm oscillations produce pressure fluctuations and large oscillating flows which create flow friction energy losses at the pump inlet and outlet. In the double-chamber pump arrangement the pressure pulses and oscillating flows are significantly reduced if the two pump chambers work in anti-phase. The pump capacity has also been doubled by having two instead of one pump diaphragm at each pump chamber. This also yields a more balanced operation of each pair of pump chamber diaphragms and a more rigidly clamped
diaphragm edge since the bending moments imposed on the thin plate structure around the edges of the oscillating diaphragms cancel out.

Another advantage of the new planar design is the broad spectrum of possible materials and pump housing fabrication techniques which can be used [2, 7, 10]. Besides the milling of metals or other materials one can also use plastics moulding techniques in low cost applications and in applications which require biocompatibility or chemical inertness and corrosion resistance [10]. A planar pump design is also suitable for planar lithography and micromachining fabrication techniques.

4. Pump and diffuser/nozzle theory

The pressure drop across a diffuser and nozzle can be written as

\[ \Delta P_d = \frac{\rho v_d^2}{2} \xi_d \]  

(1)

and

\[ \Delta P_n = \frac{\rho v_n^2}{2} \xi_n \]  

(2)

The notations are explained in Fig. 4. Here \( \rho \) is the fluid density and \( v_d \) and \( v_n \) are the fluid flow velocities in the narrowest parts of the diffuser and the nozzle, respectively. \( v_d \) and \( v_n \) are assumed to be constant across the cross sections. \( \xi_d \) and \( \xi_n \) are the pressure loss coefficients of the diffuser and the nozzle [12].

The oscillating diaphragm results in a cavity volume variation

\[ V_c = V_o \sin \omega t \]  

(3)

where

\[ V_o = K_o x_o \]  

(4)

where \( \omega = 2\pi f \), \( V_o \) is the volume variation amplitude, \( K_o \) is a constant, \( x_o \) is the center diaphragm deflection amplitude and \( f \) is the pump frequency. The net volume flow is then

\[ \phi_o - \phi_i = \frac{dV_c}{dt} = V_o \omega \cos \omega t \]  

(5)

where \( \phi_o \) and \( \phi_i \) are the momentary outlet and inlet volume flows. The pressure loss coefficients of the nozzle and diffuser are assumed to be constant throughout the pump cycle.

By using these equations and by integrating over a complete pump cycle an approximate expression for the net volume transported through the pump during a pump cycle can be obtained. The net pump volume flow (at zero pump pressure across the pump) can be written as

\[ \Phi = \frac{V_o \omega}{\pi} \left( \frac{\eta^{1/2} - 1}{\eta^{1/2} + 1} \right) = \frac{K_o x_o \omega}{\pi} \left( \frac{\eta^{1/2} - 1}{\eta^{1/2} + 1} \right) \]  

(6)

where \( \eta = \xi_n/\xi_d \), i.e. the ratio of the pressure loss coefficients of the nozzle and diffuser [12]. From the derived expression for the maximum pump flow one sees that the ratio \( \eta = \xi_n/\xi_d \) should be as large as possible in order to have a high maximum flow.

The pressure loss and volume flow equations can also be used to derive an approximate expression for the maximum pressure across one diffuser/nozzle element during a pump cycle:

\[ \Delta P_{d,\text{max}} = \frac{\rho v_o^2 A}{2A^2(1 + \eta^{1/2})^2} = \frac{\rho v_o^2 \xi_n}{2A^2(\eta^{1/2} - 1)^2} \]  

(7)

where \( A \) is the diffuser neck cross-sectional area.

The maximum volume flow in the diffuser direction, \( \phi_{d,\text{max}} \) and the nozzle direction, \( \phi_{n,\text{max}} \), during a pump cycle is

\[ \phi_{d,\text{max}} = \eta^{1/2} \phi_{n,\text{max}} = V_o \omega \frac{\eta^{1/2}}{1 + \eta^{1/2}} = \frac{\pi \eta^{1/2}}{\eta^{1/2} - 1} \Phi \]  

(8)

An approximate value of the resonance frequency of the diaphragm oscillation can be calculated by using a simplified mass-spring analogy where the elastic properties of the diaphragm represent the spring and the oscillating fluid in the pump represents the mass. This mass loading of the diaphragm is dominated by the fluid in the two diffuser/nozzle elements due to the large fluid acceleration in the narrow diffuser/nozzle elements. By using an energy analysis technique (where the losses are neglected) the total energy can be shown to oscillate between the maximum kinetic energy of the fluid in the diffuser/nozzle element and the maximum potential energy of the diaphragm. With \( P_i = P_o = 0 \), the total kinetic energy of the diffuser and nozzle elements can (see Fig. 4) be shown to be

\[ E_k = \sum \frac{m(y) v(y)^2}{2} + \sum \frac{m(z) v(z)^2}{2} \]  

(9)

where the fluid mass elements in the diffuser and a nozzle are

\[ \Delta m(y) = \rho A(y) \Delta y \]  

(10)

and
\[ \Delta m(z) = \rho A(z) \Delta z \] (11)

respectively, where

\[ \nu(y) = \phi_d \alpha A(y) \] (12)

and

\[ \nu(z) = \phi_n A(y) \] (13)

are the fluid element velocity at \( y \) and \( z \) in the diffuser and nozzle, respectively. \( \phi_d \) and \( \phi_n \) are momentary diffuser and nozzle volume flows, and \( A(y) \) and \( A(z) \) are the cross-sectional areas. In the case of flat diffuser (nozzle) walls one has

\[ A(y) = b \left( d + (D - d) \frac{y}{L} \right) \] (14)

and

\[ A(z) = b \left( d + (D - d) \frac{z}{L} \right) \] (15)

where \( d \) is the diffuser inlet width, \( D \) is the diffuser outlet width, \( b \) is the diffuser depth and \( L \) is the diffuser length.

The kinetic energy is then

\[ E_k = \int_0^L \frac{\phi^2 d}{2A(y)^2} \rho A(y) dy + \int_0^L \frac{\phi^2 d}{2A(z)^2} \rho A(z) dz = \frac{\rho}{2b} \frac{L}{(D - d) \ln \frac{D}{d}(\eta + 1)} \phi_n^2 \] (16)

which yields the maximum kinetic energy as

\[ E_{k,\text{max}} = \frac{\rho L}{2b(D - d)} \ln \frac{D}{d}(\eta + 1) \phi_n^2 \max \]

\[ \rho L (K_n x_0 \omega_0)^2 (\eta + 1) \ln \frac{D}{d} \]

\[ = \frac{\rho L (K_n x_0 \omega_0)^2 (\eta + 1) \ln \frac{D}{d}}{2b(1 + \eta^{1/2})^2 (D - d)} \] (17)

The potential energy of the diaphragm can be written as

\[ E_p = \frac{K_p K_n}{2} \frac{x^2}{\pi} \] (18)

where \( K_p \) is a constant, the diaphragm 'spring constant', defined from

\[ P = \frac{K_p x}{\pi} \]

where \( x \) is the diaphragm centre deflection as a result of the pressure, \( P \), acting on the diaphragm. The maximum diaphragm potential energy is then

\[ E_p = \frac{K_p K_n}{2} \frac{x^2}{\pi} \] (19)

Since the total energies of the oscillating fluid and diaphragm are equal to the maximum kinetic energy and the maximum potential energy

\[ E_{k,\text{max}} = E_{p,\text{max}} \] (20)

Thus the resonance frequency \( f_o \) is

\[ f_o = \frac{1}{2 \pi} \left[ \frac{K_n (1 + \eta^{1/2})^2 b (D - d)}{\rho K_s (\eta + 1) L \ln \frac{D}{d}} \right]^{1/2} \] (21)

An important parameter in fluid systems is the Reynolds number, \( Re \), which for a stationary flow situation can be used to determine whether the flow is laminar or turbulent. For pipe flow systems \( Re < 2300 \) indicates that the flow is laminar and for \( Re > 2300 \) that the flow is turbulent [14]. The maximum Reynolds number is found in the narrowest flow path of the pump, i.e. the diffuser/nozzle neck, and in the case of a quadratic neck cross section, \( b = d \), can be written as

\[ Re = \frac{\nu d}{\gamma A_d \gamma b d \gamma} = \frac{\phi}{\frac{d}{\gamma} A_d \gamma b d \gamma} \] (22)

where \( \nu \) is the mean flow velocity, \( \phi \) is the volume flow in the cross section and \( \gamma \) is the kinematic viscosity \((\gamma = 1.01 \times 10^{-6} \text{ m}^2/\text{s for water})\).

This yields the maximum momentary Reynolds number as

\[ Re_{\text{max}} = \frac{\phi_{\text{d, max}}}{b \gamma} = \frac{1}{b \gamma (\eta^{1/2} - 1)} \frac{\pi \eta^{1/2}}{\phi} \] (23)

This expression for the maximum Reynolds number in the pump during the pump cycle can be used as an indicative and for deciding if the flow in the diffuser is laminar or turbulent.

These analyses are based on a single chamber with a single diaphragm, but the equations can easily be modified to describe the double-chamber and four-diaphragm design by doubling the chamber volume amplitude and the chamber net flow.

5. Results

The pump characteristics were measured in a set-up where silicon rubber tubes with an inside diameter of 2 mm were connected to the pump inlet and outlet. The pump system was completely filled with water. The diaphragms were excited with a square-wave voltage which resulted in a sinusoidal diaphragm vibration due to the oscillating character of the fluid in the pump. The diaphragm vibration was measured using a fibre
optical detection system. The pump flow was measured by continuously weighing the output fluid flow in known time intervals. The pump pressure, defined as \( P_0 - P_1 \) in Fig. 4, was achieved by letting the output fluid level be at a variable, measured elevation above the input fluid level. The excitation frequency was normally set to the resonance frequency of the pump for water. The parallel double-chamber pump was designed for a push–pull anti-phase chamber excitation, but, for comparison and evaluation purposes, pump measurements were also conducted with the pump chambers excited in the in-phase chamber excitation mode.

In Fig. 5 the pressure–flow characteristics of the pump excited in the double-chamber anti-phase mode are shown for different excitation voltages. The diaphragm amplitudes at zero pump pressure are given for each excitation voltage. The resonance frequency of the anti-phase mode was 540 Hz and the \( Q \) factor was about 5.

The maximum pump flows from Fig. 5 are plotted in Fig. 6(a) as a function of the diaphragm amplitude. In the diagram the calculated straight line dependency based on Eq. (6) is also shown. The measurements show that the diaphragm amplitude must be larger than 0.76 \( \mu \)m for diffuser pump flow action. The reason for this is probably that the flow directing property of the diffuser/nozzle element is lost for low pump flow velocities. This assumption is supported by the stationary measurements below, which show that \( \eta = \xi_1/\xi_2 \leq 1 \) for low diffuser/nozzle element pressure drops. It is interesting to note that the measurements show a pump action for a diaphragm amplitude as low as 0.76 \( \mu \)m and a volume flow of 0.05 ml/min per pump chamber. According to Eq. (23) this yields an equivalent maximum Reynolds number, \( Re_{max} = 29 \), which is much below the critical value of 2300 which is used as a laminar flow criterion. In fact, all the pump flow values below 7.8 ml/min result in calculated Reynolds numbers which never exceed 2300 throughout the pump cycle. However, the non-stationary oscillating nature of the flow in the non-uniform cross sections of the diffuser/nozzle elements makes it very difficult to predict the type of flow in the elements.

The pump measurement data in the diagram in Fig. 5 have also been used to plot the maximum pump pressure as a function of the diaphragm amplitude in Fig. 6(b).

Both the maximum flow and maximum pump pressure dependences of the excitation frequency at a constant excitation voltage of 80 V p–p were measured and plotted in Fig. 7, which shows that the optimum excitation frequencies for the maximum pump pressure and the maximum pump flow are the same (560 Hz) and slightly higher than the resonance frequency of the pump (540 Hz). In the case of the volume flow, the reason for this is the linear relation between the maximum net volume flow and both the diaphragm amplitude and the excitation frequency \( \Phi \sim x_0(\omega)\omega \), which shows that the volume flow will have a peak value at a higher frequency than the amplitude diaphragm amplitude.

The pressure–flow characteristics of the pump operated in the in-phase pump chamber mode are given

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Fig. 5. Pressure–flow characteristics of the two-chamber pump for water in anti-phase operation for different excitation voltages (and diaphragm amplitudes at zero pump pressure). The diaphragm excitation frequency was 540 Hz.

Fig. 6. (a) Maximum flow (at zero pump pressure) and (b) maximum pump pressure (at zero pump flow) as a function of the diaphragm amplitude for anti-phase operation. The solid straight line is the theoretically determined maximum flow.
(18), respectively, for a brass pump diaphragm with a piezoelectric PZT disc were calculated using a FEM program (ANSYS) as $K_r = 5.0 \times 10^{-5} \text{ m}^2 \text{ Pa}$ and $K_p = 2.6 \times 10^{10} \text{ Pa/m}$. The lower value of the measured resonance frequency of 540 Hz is probably due to the fluid masses in the tubes which participate in the fluid oscillation. This fluid tube mass is neglected in the theoretical calculations, which therefore yields a higher estimated resonance pump frequency. Furthermore, the velocity profile in a cross section of the diffuser/nozzle element is not flat, as assumed in the simplified analysis. The flow velocities at the walls and especially at the corners of the walls are much lower than the bulk velocity in a diffuser/nozzle element cross section, indicating a lower value of the resonance frequency than that calculated.

The flow rectifying properties of the pump were investigated by forcing water through the pump, without diaphragm excitation while measuring the pressure drop and volume flow. The parallel double-chamber arrangement means that the two pairs of serially mounted diffusers of each pump chamber are in parallel, which means that the pressure drop and flow through each diffuser/nozzle element can be assumed to be half the total pump pressure and half the total volume flow. The stationary pressure–flow measurements are plotted in Fig. 9. The pressure–flow data together with Eqs. (1) and (2) were used to calculate and plot the diffuser and nozzle pressure loss coefficients $\xi_d$, $\xi_n$ and the ratio $\eta = \xi_n / \xi_d$ in Fig. 10. Based on the maximum net pump flow in Fig. 7 of 16 ml/min and Eq. (7), the maximum pressure drop for a diffuser/nozzle element during a pump cycle could be calculated to be 101 kPa (with $\xi_d = 1.6$ and $\eta = 2$), which demonstrates that most of the chamber pressure oscillation of the pump flow measurements takes place in the measured range of the stationary measurements of Fig. 10. The measurements show that the pressure loss coefficient ratio, $\eta$, is less than unity for low pressure drops (below 0.3

in Fig. 8. The measurements show that, at the same excitation voltage of 80 V, the anti-phase operation mode yields a maximum flow value which is more than twice as high as the in-phase operation mode, and a maximum pump pressure about three times as high. The explanation for this is that the tube fluid mass loading is lower in the anti-phase operation mode than in the in-phase operation mode due to the internal pump flow compensation of the anti-phase operated pump. This gives a higher diaphragm amplitude as well as a higher resonance frequency of the anti-phase operated pump. The resonance frequency was 540 Hz for anti-phase and 470 Hz for in-phase operation. The diaphragm amplitude was 5.1 $\mu$m for anti-phase and 2.4 $\mu$m for in-phase operation at an excitation voltage of 80 V.

By using Eq. (21) a value of the resonance frequency of 735 Hz could be calculated. The calculation is based on our design with two diaphragms per pump chamber and an assumed pressure loss coefficient ratio $\eta$ equal to two. The constants $K_r$ and $K_p$, from Eqs. (4) and
7. Conclusions

We have fabricated and tested a planar double-chamber pump in brass based on the new valve-less diffuser pump principle. Each chamber has two piezoelectrically driven diaphragms with a diameter of 13 mm.

It has been demonstrated that the planar parallel double-chamber configuration excited in the anti-phase mode was twice as effective as the in-phase operation mode in terms of pump flow and pump pressure. Reasonably good agreement between approximate theoretical analyses and measurements was achieved. The maximum pump flow was $16 \text{ ml/min}$ and the maximum pump pressure was $1.7 \text{ m } \text{H}_2\text{O}$ when the diaphragms were excited at the pump resonance frequency of 540 Hz. The stationary flow rectifying properties of the diffuser/nozzle element with rectangular cross section were verified in pressure–flow measurements of the pump structure.

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References