Micropumps

The main applications for which micropumps have been developed are drug delivery and taking and transporting of samples for analysis. It is much easier to build a micropump than a microvalve, and much more publications exist on micropumps than on microvalves. The problem with micropumps is that up to now there is no large market for them. Many tasks which could be done by micropumps can be done by even easier devices. For example, drug delivery can be done by an infusion bottle hanged up.

Typical specifications for micropumps are the maximum flow rate and counter pressure, accuracy of dosing and flow rate, delivery of gases and liquids, self priming, tolerance against particles and bubbles in the fluid, life time, small dimensions and weight (with power supply and electronics), small energy consumption, small temperature change and pulsation of the fluid, chemical and biological inertness, and low price.

Most micropumps are reciprocating pumps, i.e., an actuator is continuously reducing and enlarging the volume of a pump chamber. Passive valves at the inlet and the outlet cause the in and out flow to go only into one direction. Figure 173 shows the principle of such a pump. The pressure at the inlet p_e is smaller than the one at the outlet p_a and both may be different than the reference pressure p_R (typically from the environment), which is acting on the backside of the actuator.

The performance of a micropump certainly is a function of its actuator. There are only a few actuator principles such as the lateral capacitive force (cf. page 132), the evaporation of a liquid in a closed volume (cf. page 171), and an external pneumatic or hydraulic drive for which the force is not a function of the stroke. Nearly all other actuators show a characteristic curve which is similar to the one of a piezo or bimaterial actuator. For example, compare the characteristic curves in Fig. 125a (page 169), Fig. 123b (page 166), and Fig. 101 (page 140) or Fig. 118 (page 162). Therefore, the following discussion assumes a piezo actuator.

For the application of an actuator for a micropump, it is best to draw the characteristic curve as the generated pressure difference Δp_A over the volume displacement ΔV_A as shown in Fig. 174. The characteristic curve is defined by the maximum pressure difference Δp_{max} which is achieved if the actuator is hindered from deflecting and the maximum volume displacement ΔV_{max} which is

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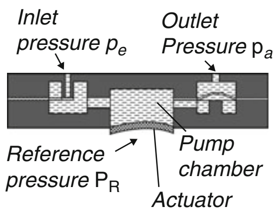


Fig. 173 Schematic cross-section of a reciprocating micropump

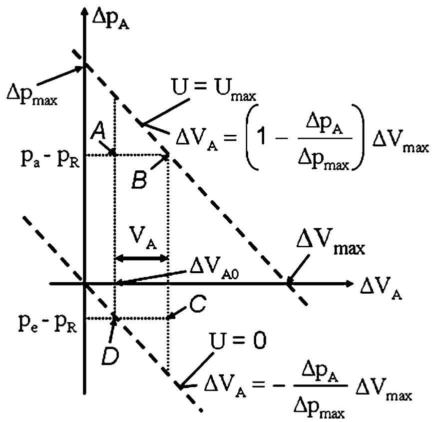


Fig. 174 Characteristic curve of a typical micropump actuator

obtained when there are no outer forces acting. The characteristic curve is described by the following equation:

$$\Delta p_A = \Delta p_{max}\left(1 - \frac{\Delta V_A}{\Delta V_{max}}\right) \Rightarrow \Delta V_A = \Delta V_{max}\left(1 - \frac{\Delta p_A}{\Delta p_{max}}\right). \tag{391}$$

The above equation is right when the maximum voltage U_{max} is applied to the piezo. If the piezo is short-circuited, the curve is shifted such that it touches the origin, and the characteristic curve is described by:

$$\Delta p_A = -\frac{\Delta p_{max}}{\Delta V_{max}}\Delta V_A \quad \text{or} \quad \Delta V_A = -\frac{\Delta V_{max}}{\Delta p_{max}}\Delta p_A. \tag{392}$$

When the actuator is powered, it raises the pressure in the pump chamber until it becomes larger than the pressure at the outlet (*A* in Fig. 174). At this point of the pump cycle, the actuator overcomes the pressure difference between the outlet and the reference pressure ($p_a - p_R$). The outlet valve opens and the actuator pushes fluid out through that valve. While doing so, the pressure which can be generated by the actuator is reducing with the volume which has been displaced, until the actuator is no longer able to overcome the pressure difference (*B*). Then the actuator is moving back (discharged if it is a piezo) and the lower characteristic curve is valid. The pressure in the pump chamber by the actuator now can be made smaller than the inlet pressure. Therefore, the outlet valve is closing, the inlet valve opens and the pressure drops a bit below the pressure at the inlet. At this point of the pump cycle (*C*), the pressure acting on the actuator is the difference between inlet pressure and reference pressure ($p_e - p_R$). Fluid is taken in through the inlet valve until the actuator can no longer provide enough underpressure (*D*). Then, a new pump cycle is started by powering the actuator again.

The volume V_A which is pumped during one cycle is found in Fig. 174 as the distance between the vertical dotted lines. It is calculated from the difference of the displaced volume ΔV_A at *B* [(391) with $\Delta p_A = p_a - p_R$] and at *D* [(392) with $\Delta p_A = p_e - p_R$]:

$$\begin{aligned} V_A &= \Delta V_{max}\left(1 - \frac{p_a - p_R}{\Delta p_{max}}\right) - \Delta V_{max}\left(-\frac{p_e - p_R}{\Delta p_{max}}\right) \\ &= \Delta V_{max}\left(1 - \frac{p_a - p_e}{\Delta p_{max}}\right). \end{aligned} \tag{393}$$

The displaced volume ΔV_A at *D* in Fig. 174 is denoted here as ΔV_{A0} for later use in this chapter:

$$\Delta V_{A0} := \Delta V_{max}\left(-\frac{p_e - p_R}{\Delta p_{max}}\right). \tag{394}$$

The pump cycle between the characteristic curves of the actuator and the volume displaced per pump cycle do not contain the information how long a pump cycle lasts. This is a function of the flow through the valves. The flow Φ_a through the outlet valve with flow resistance R_{fl} is calculated with (355) (page 207). The pressure difference Δp_V over the valve is with (391):

$$\Delta p_V = p_R + \Delta p_A - p_a = p_R - p_a + \Delta p_{max}\left(1 - \frac{\Delta V_A}{\Delta V_{max}}\right) \tag{395}$$

$$\Rightarrow \Phi_a = \frac{-\Delta p_V}{R_{fl}} = \frac{p_R - p_a + \Delta p_{max}\left(1 - \frac{\Delta V_A}{\Delta V_{max}}\right)}{R_{fl}} = \frac{\partial \Delta V_A}{\partial t}. \tag{396}$$

The flow through the outlet valve as described by the above equation is equal to the derivative of the volume displacement of the actuator. As a consequence, the volume displacement as a function of time can be calculated by separation of variables and integration of the above equation:

$$\Rightarrow \int_0^t \frac{dt'}{R_{fl}} = \int_{\Delta V_{A0}}^{\Delta V_A} \frac{d\Delta V_A{}'}{p_R - p_a + \Delta p_{max} - \frac{\Delta p_{max}}{\Delta V_{max}} \Delta V_A{}'} \tag{397}$$

$$\Rightarrow \frac{t}{R_{fl}} = -\frac{\Delta V_{max}}{\Delta p_{max}} \ln\left(\frac{p_R - p_a + \Delta p_{max} - \frac{\Delta p_{max}}{\Delta V_{max}} \Delta V_A}{p_R - p_a + \Delta p_{max} - \frac{\Delta p_{max}}{\Delta V_{max}} \Delta V_{A0}}\right) \tag{398}$$

$$\Rightarrow \Delta V_A = \left(\frac{p_R - p_a}{\Delta p_{max}} + 1\right)\Delta V_{max} - \left(\left(\frac{p_R - p_a}{\Delta p_{max}} + 1\right)\Delta V_{max} - \Delta V_{A0}\right) e^{-\frac{\Delta p_{max}}{\Delta V_{max}} \frac{t}{R_{fl}}} \tag{399}$$

In the above equations, it was assumed that the flow resistance R_{fl} of the valve is not a function of time. In general this is not true, but the valves of a pump should be designed such that the stroke for most of time is large to allow a flow which is limited only by the cross-section of the feed channels (cf. Fig. 157 on page 208). Thus, a constant flow resistance of the valves appears to be a suitable approximation.

Inserting (394) into (399) yields:

$$\Delta V_A = \Delta V_{max}\left[\left(\frac{p_R - p_a}{\Delta p_{max}} + 1\right) - \left(\frac{p_e - p_a}{\Delta p_{max}} + 1\right) e^{-\frac{\Delta p_{max}}{\Delta V_{max}} \frac{t}{R_{fl}}}\right] \tag{400}$$

The flow through the outlet valve is now obtained by either inserting the above equation into (396) or by differentiating the above equation with respect to time:

$$\Phi_a = \frac{\Delta p_{max} - \Delta p_P}{R_{fl}} e^{-\frac{\Delta p_{max}}{\Delta V_{max}} \frac{t}{R_{fl}}} \tag{401}$$

In the above equation, the pressure drop over the pump was denoted as $\Delta p_P = p_a - p_e$.

The flow through the outlet valve as a function of time and the radius R_Z of the feed channels and the valve seat calculated with (401) is shown in Fig. 175. A pump actuator was assumed which is able to generate a maximum pressure difference Δp_{max} of 20 kPa and a maximum volume displacement ΔV_{max} of 100 nL. The pressure difference over the pump Δp_P was 10 kPa and water (viscosity = 1 mPa s) was used as the fluid. The flow resistance R_{fl} of the valve was calculated with (355) (page 207) for a passive valve with circular cross-section and feed channels and a large stroke (the first term in the large parenthesis in the denominator is neglected). The length of the feed channels was assumed to be 500 µm.

In Fig. 175, the flow through the outlet valve of the pump is increasing with the size of the valve and the feed channels. This is due to the decreasing friction of the fluid in the feed channels. Thus, it appears as if it would be best to design as large feed channels as possible. This is not true, because the dead volume of the valves has a major effect and has not been taken into account in the above calculation.

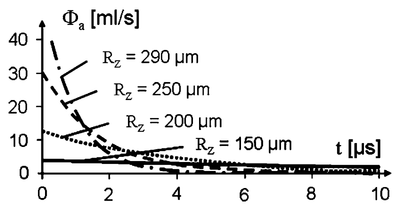


Fig. 175 Flow through the outlet valve of a micropump as a function of time and the radius of the feed channel. The dead volume of the valve is not taken into account

The volume delivered with each pump stroke is not the volume V_A shown in Fig. 174, but the actuator needs to displace the dead volume of the passive valves before fluid is leaving the outlet valve.

A certain pressure difference is necessary to open the outlet valve. Therefore, the point *A* in Fig. 176 is a bit above the pressure ($p_a - p_R$) which is necessary to overcome the pressure at the outlet. This is not taken into account here. Instead Fig. 176 shows at *A* the effect of stiction of a valve what may occur.

Then fluid is displaced out of the pump chamber and leaves through the outlet valve. When point *B* is reached, no more fluid can be displaced by the actuator and it is necessary to discharge the piezo and switch to the lower characteristic curve. As a consequence, the outlet valve is closing and its dead volume V_a is flowing backward.

The pressure in the pump chamber is decreasing now until it is below the pressure at the inlet ($p_e - p_R$) and is able to open the inlet valve (*C* in Fig. 176). The actuator is moving back then until it is no longer able to produce a pressure which is less than the pressure at the inlet (*D*). The piezo is charged again and the inlet valve is closing. As a result, the dead volume V_e of the inlet valve is displaced by the actuator in backward direction. After that the next pumping cycle is starting.

The volume V_F which is delivered by one pump cycle is the distance V_A between the dotted lines in Fig. 174 (page 230) [(calculated with (393)] minus the dead volumes of inlet V_e and outlet V_a valve:

$$V_F = V_A - V_e - V_a = \Delta V_{max}\left(1 - \frac{p_a - p_e}{\Delta p_{max}}\right) - V_e - V_a \tag{402}$$

The volume displaced by the actuator as a function of time can now be calculated with (399), if the dead volume of the inlet valve V_e is added to ΔV_{A0} in (394) (page 231):

$$\Delta V_A = \Delta V_{max}\left[\left(\frac{p_R - p_a}{\Delta p_{max}} + 1\right) - \left(\frac{p_e - p_a}{\Delta p_{max}} + 1 - \frac{V_e}{\Delta V_{max}}\right) e^{-\frac{\Delta p_{max}}{\Delta V_{max}}\frac{t}{R_{fl}}}\right] \tag{403}$$

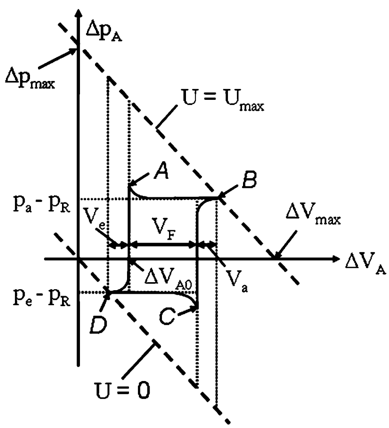


Fig. 176 Characteristic curve of a typical micropump actuator

The flow Φ_a through the outlet valve as a function of time is obtained when the above equation is inserted into (396) or by differentiating the above equation with respect to time:

$$\Phi_a = \frac{(1 - (V_e/\Delta V_{max}))\Delta p_{max} - \Delta p_P}{R_{fl}} e^{-\frac{\Delta p_{max}}{\Delta V_{max}} \frac{t}{R_{fl}}} \tag{404}$$

The dead volume of the inlet valve was calculated according to (388) (page 222), and the stroke was assumed to be as large as half of the radius of the feed channel:

$$V_e = \pi\, R_Z^2\, H = \frac{1}{2}\pi\, R_Z^3. \tag{405}$$

The flow Φ_a through the outlet valve calculated with (404) and (405) is shown in Fig. 177. The same parameters were used as for Fig. 175. The comparison of these two figures shows that the volume flow through the outlet valve is rising with the radius of the valve seat and the feed channels. However, if the dead volume of the valves is taken into account, there is an optimum radius. If this optimum radius is exceeded, the flow through the outlet valve becomes smaller again, although the time which is required to empty the pump chamber gets shorter. The reason for this is that the pressure difference which is available to push the fluid through the outlet valve is smaller for a larger dead volume and the volume which can be delivered by one pump stroke is reduced also. This is recognized in Fig. 176: the position of the point *A* is moving to the right when the dead volume of the inlet valve becomes larger.

On the other hand, for small radii of feed channels and valve seats, the volume flow Φ_a through the outlet valve is rising with increasing radii because the flow resistance R_{fl} is decreasing.

The flow rate of a reciprocating pump consists of repeated strokes of fluid volumes leaving the outlet. This is shown in Fig. 178 for different pumping frequencies. At a low frequency (Fig. 178a), nearly all of the volume in the actuator is delivered before the actuator is switched to its lower characteristic curve at the time t_f. Then the outlet valve is closing and its dead volume is flowing in backward direction during the switching time of the valve.

Next the intake stroke starts, the inlet valve opens, and fluid is sucked into the pump chamber. The actuator is switched to its upper characteristic curve and the dead volume of the inlet valve is flowing in backward direction. No flow is observed at the outlet during this time.

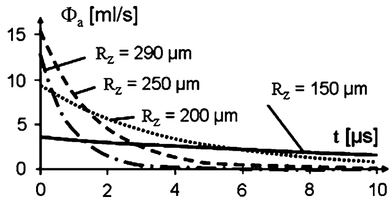


Fig. 177 Flow through the outlet valve of a micropump as a function of time and the radius of the feed channel calculated with (404) and for the dead volumes of the valves (405)

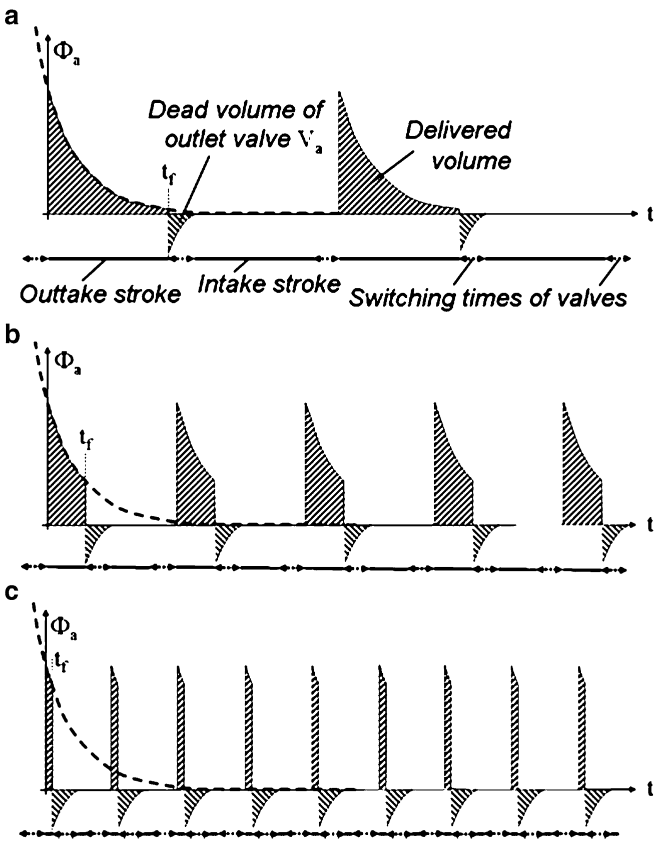


Fig. 178 Flow Φ_a through the outlet valve of a pump as a function of time for (**a**) a low, (**b**) a medium, and (**c**) a high frequency of the pump actuator. The *dashed area* represents the delivered volume

A new pump cycle starts when the outlet valve opens again and fluid is pressed out of the pump chamber.

When the pump frequency is low (cf. Fig. 178a), most of the time there is only a small or even no volume flow at the outlet of the pump. At a somewhat higher frequency (Fig. 178b), only the part with a large flow through the outlet valve is used and the part with a smaller flow rate is forgone to start the next pump cycle earlier. As a consequence, the mean volume flow generated by the pump (the integral over Φ_a divided by the time of a pump cycle) is larger than at a lower frequency.

If the frequency is enhanced even more, the volume delivered in a pump cycle in forward direction is not much more than flowing backward, and the actuator is only moving the dead volume of the valves back and forth. As a result, there is an optimum frequency at which the pump delivers the maximum possible flow.

The volume V_F delivered by one pump stroke is the displaced volume ΔV_A at the time t_F when the actuator is switched from the upper characteristic curve to the lower one (*B* in Fig. 176) minus the dead volume V_a of the outlet valve. This volume V_F delivered by one pump stroke needs to be multiplied with the pump frequency f_P to obtain the flow rate Φ_P of the pump. With (403) it is obtained:

$$\Phi_P = f_P V_F = f_P[\Delta V_A(t_F) - \Delta V_A(0) - V_a] \tag{406}$$

$$\Rightarrow \Phi_P = f_P \left\{ \Delta V_{max} \left(1 - \frac{\Delta p_P}{\Delta p_{max}} - \frac{V_e}{\Delta V_{max}} \right) \left[1 - e^{-\frac{\Delta p_{max}}{\Delta V_{max}} \frac{t_f}{R_{fl}}} \right] - V_a \right\} \tag{407}$$

The time t_f when the actuator is switched from one characteristic curve to the other can be estimated from the switching time of the valves and the frequency of the pump actuator. If it is assumed that the switching time t_V of inlet and outlet valve are the same and the duration of intake and outtake stroke are equal, it is obtained:

$$t_f = \frac{1}{2} \frac{1}{f_P} - t_V \tag{408}$$

The switching times t_V of inlet and outlet valves are estimated by (390) on page 223. The dead volume of the valves is taken from (405) and the flow resistance R_{fl} of the valves was calculated with (355) (page 207) for a passive valve with circular cross-section and feed channels and a large stroke [the first term in the large parenthesis in the denominator of (355) is neglected]. The pressure difference over the valves during the back flow was estimated by the maximum pressure Δp_{max} which the actuator is able to generate:

$$R_{fl,Z} = 8\,\eta \frac{4\,L_Z}{D_{h,z}^2\,A_Z} = \frac{8\,\eta\,L_Z}{\pi\,R_z^4} \quad \text{and} \quad t_V = \frac{R_{fl,Z}\,V_V}{\Delta p_V} = \frac{4\,\eta\,L_Z}{R_Z\,\Delta p_{max}} \tag{409}$$

By inserting (408) and (409) into (407), the flow rate of a pump was calculated as a function of the driving frequency, and the radii of the feed channels and the valve seats. For the other parameters, the same values as for the calculation shown in Fig. 175 were used. In Fig. 179, the result of this calculation is shown.

For small frequencies, the flow rate is increasing proportional to the frequency because the volume displaced by each pump stroke is delivered more often. At higher frequencies, the volume V_F delivered by each pump stroke is decreasing because the actuator is switched to the lower characteristic curve before all the volume which could be displaced by the actuator has left the outlet valve and because V_F is no longer large compared with the dead volume of the valves. However, the flow rate is

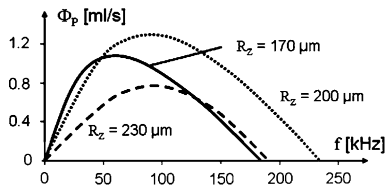


Fig. 179 Flow Φ_P of a micropump as a function of actuator frequency f and the radius R_Z of feed channels and valve seats as calculated with (407)

decreasing with increasing frequency at very high frequencies (The scale is in kHz.). It is virtually impossible for a pump actuator to generate such high frequencies. In the literature, there was reported on a micropump which started to pump in reverse direction when the resonance frequency of the moving fluid and the valves was approached [64]. Here the effect of the acceleration of the fluid and the valve bodies and resonance effects have not been taken into account. Such effects occur only at large flow velocities which rarely are found at micropumps.

It is also seen in Fig. 179 that the slope of the curve at small frequencies is a function of the radius of feed channels and the valve seats. The effect of the size of the valves and feed channels is seen better when the pump flow calculated with (407) is drawn as a function of the radius of these components (cf. Fig. 180).

In Fig. 180a, the pump flow is shown for different actuator frequencies. It is clearly seen that the radius R_Z at which the largest flow is achieved is a function of frequency. If the radius is a bit larger than the optimum value, not much of the possible flow is lost. However, if the size of the valves and feed channels is only a bit too small, this may cause a remarkable loss in flow.

It is also seen in that figure that the maximum radius $R_{Z,max}$ at which a flow can be generated is not a function of frequency. Figure 180b shows that $R_{Z,max}$ is a function of the counter pressure Δp_P. For a given radius of the feed channels, there is maximum counter pressure $\Delta p_{P,m}$ which limits the range of operation of the pump. At $R_{Z,max}$, the flow resistance R_{fl} of the feed channels and the valves is very small and the exponential function in (407) can be neglected compared with 1. At $R_{Z,max}$, the flow calculated with (407) is zero:

$$0 = f_P \left\{ \Delta V_{max} \left(1 - \frac{\Delta p_P}{\Delta p_{max}} \right) - V_e - V_a \right\} \Rightarrow \Delta p_{P,m} = \left(1 - \frac{V_e + V_a}{\Delta V_{max}} \right) \Delta p_{max}. \quad (410)$$

The above equation means that the maximum pressure Δp_P which can be generated by a pump is only a function of the maximum pressure Δp_{max} generated by the actuator, the ratio of the sum of the dead volumes of inlet V_e and outlet V_a valve

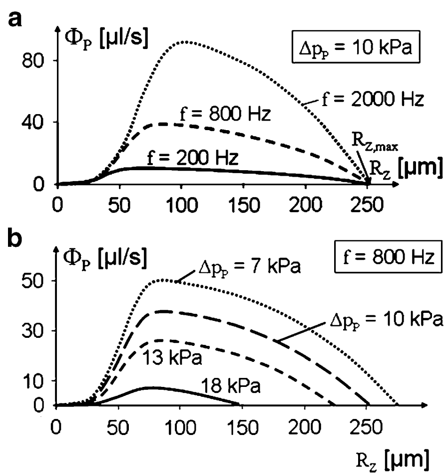


Fig. 180 Flow Φ_P of a micropump as a function of the radius R_Z of feed channels and valve seats and (**a**) the actuator frequency f and (**b**) the counter pressure Δp_P, respectively, calculated with (407)

and the maximum volume ΔV_{max} which can be displaced by the actuator. The maximum pressure of a pump is a function of the size of the valves, because their dead volume is a function of that size. If (405) is employed to calculate the dead volume, the maximum pressure $\Delta p_{P,max}$ of a pump as a function of the radius R_Z of the valve seat is:

$$\Delta p_{P,m} = \left(1 - \frac{\pi\, R_Z^3}{\Delta V_{max}}\right) \Delta p_{max}. \tag{411}$$

The maximum pressure $\Delta p_{P,m}$ generated by the pump with the parameters used above was calculated with the above equation and drawn as a function of R_Z in Fig. 181. It is clearly seen that the largest pressure can be generated with a pump which has very small valves. The smaller the valves are the more the maximum pressure $\Delta p_{P,m}$ generated by the pump approaches the maximum pressure Δp_{max} generated by the actuator. However, small valves will result in a small flow (cf. Fig. 180). Therefore, a compromise needs to be made between a large flow and a large pressure generated.

The flow generated by a pump as a function of the counter pressure is called the characteristic curve of the pump. This curve is obtained when (407) is plotted as a function of the counter pressure. Figure 182 shows the characteristic curve of a pump with the parameters used above for different radii of feed channels and valve seats. The characteristic curve is a straight line between the maximum flow $\Phi_{P,m}$ and the maximum pressure $\Delta p_{P,m}$ which can be generated by the pump.

Obviously, the flow is rising for small valves with their size while the maximum pressure which can be achieved is nearly not changing (see also Fig. 181). In this example, the optimum radius is reached at approximately 65 μm. When the radius R_Z exceeds this value, the characteristic curve is shifting left and down nearly

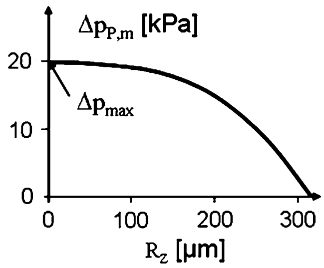


Fig. 181 Maximum pressure $\Delta p_{P,m}$ which can be generated with a pump as a function of the radius R_Z of feed channels and valve seats

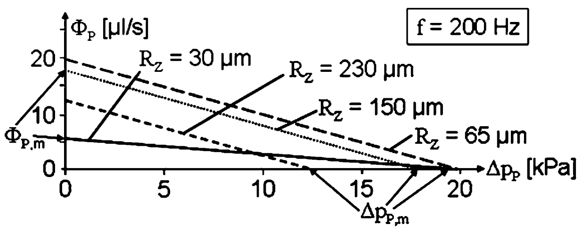


Fig. 182 Characteristic curve of a pump as a function of the radius R_Z of feed channels and valve seats

parallel. Thus, maximum flow rate and maximum pressure both are reducing when the radii R_Z of the feed channels and the valves are increasing.

If the dead volume and the flow resistance of the valves would have no effect on maximum flow rate and maximum pressure $\Delta p_{P,m}$ which can be generated by the pump, the maximum flow $\Phi_{P,m}$ would be equal to the frequency times the maximum pressure which can be generated by the actuator (20 μL/s in Fig. 182) and the maximum pressure generated by the pump would be equal to the maximum pressure Δp_{max} of the actuator (20 kPa in Fig. 182). It is seen in the figure that at the optimum radius these values are nearly reached (19.8 μL/s and 19.8 kPa, respectively). However, even larger flow rates can be achieved, if active valves are employed and the flow resistance of the valve is reduced by designing them larger. The disadvantage of this concept is that three actuators and a suitable electronic control are required. An additional advantage is that a pump with active valves can pump also in reverse direction.

The *fluidic power of a pump* P_P is the product of the flow rate Φ_P and the counter pressure Δp_P. Thus, the fluidic power is calculated as the product of (407) with the counter pressure. Since the characteristic curve is a linear function of the pressure, the result of multiplying with the pressure is a parabola and the maximum $P_{P,m}$ of that parabola is a forth of the product of maximum counter pressure $\Delta p_{P,m}$ and maximum flow rate $\Phi_{P,m}$:

$$p_{P,max} = \frac{1}{4}\Phi_{P,m}\,\Delta p_{p,m} \tag{412}$$

The fluidic power for the graphs shown in Fig. 182 is drawn in Fig. 183. The input power of the piezo actuator is not much affected by the counter pressure and the flow generated (cf. page 150). Therefore, the power output shown in Fig. 183 is nearly proportional to the efficiency of the pump. If micropumps are to be designed for portable or implantable devices which are powered by batteries, this may be an important issue. Since the fluidic power P_P is a parabolic function of the counter pressure, fluidic power and efficiency of a pump are maximum at half of the maximum counter pressure $P_{P,m}$. Therefore, when energy consumption of a pump is a concern, it should be designed such that the typical counter pressure is half of the maximum counter pressure.

When the radii R_Z of the feed channels and the valve seats are increased, the maximum of the fluidic power is increasing, while the optimum counter pressure is nearly not decreasing. When the maximum fluidic power which can be achieved by

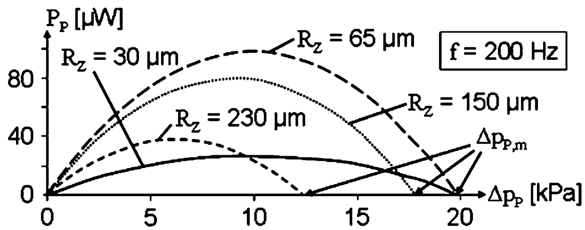


Fig. 183 Fluidic power of a micropump as a function of the counter pressure and the radii of feed channels and the valve seats

increasing the radii R_Z is exceeded, the optimum counter pressure and the maximum fluidic power are decreasing.

When a pump is employed for a dosing application, it is desirable that the flow rate is not or only a week function of the counter pressure. This can be achieved by limiting the maximum stroke of the actuator (designing a very shallow pump chamber) to much less than the maximum stroke which the actuator could perform without that restriction. As a consequence, the pressure generated by the actuator during the stroke is nearly constant.

Even better is employing an actuator for which the force generated is not a function of the stroke such as the lateral capacitive force (cf. page 132), evaporation of a liquid in a closed volume (cf. page 171), and an external pneumatic or hydraulic drive.

Another option is to design two small *pumps working in series* instead of a larger pump with the same volume. The first of the two pumps needs to be driven such that its outtake stroke is over before the intake of the second pump starts. If both pumps are driven in phase, they behave similar as a single pump with a larger pump chamber.

If the two pumps are not driven in phase, each of them needs to overcome a smaller counter pressure and they have a smaller maximum flow rate, because their pump chambers are smaller than the chamber of a single pump with the same volume as both pumps together. The resulting characteristic curves are shown in Fig. 184a. If the maximum volume V_{max} displaced by the actuator and the dead volumes of inlet V_e and outlet V_a valves are all reduced by the same factor, e.g. 2, the maximum flow $\Phi_{P,m}$ is also reduced by that factor (cf. 407) and the maximum pressure generated by such a small pump is not changing (cf. 410). If two pumps are working in series and are driven with a phase shift, the maximum pressure generated by both pumps together is doubled. As shown in Fig. 184a, the slope of the characteristic curve of the coupled pumps is significantly reduced compared with the larger single pump and a change of the counter pressure Δp_P has a much smaller effect on the flow rate Φ_P. The flow rate can be adjusted then by choosing the driving frequency of the actuator.

Another interesting effect of coupled pumps driven not in phase is that the maximum efficiency of pumping is shifted to a larger counter pressure. This is

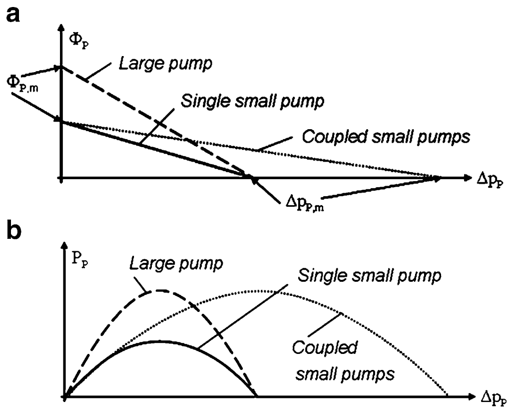


Fig. 184 (**a**) Characteristic curve of a pump, a small pump, and two small pumps in series driven with a phase shift and (**b**) fluidic power of these pumps

shown in Fig. 184b. Since the maximum fluidic power is a fourth of the product of maximum flow $\Phi_{P,m}$ and maximum counter pressure $\Delta p_{P,m}$ (cf. 412), the maximum fluidic power of two small coupled pumps with half of the maximum flow is the same as for a larger pump. But the maximum is at a larger counter pressure and a certain change of the counter pressure does not affect the fluidic power so much. Therefore, coupling two or even more pumps may save battery power in certain applications.

Leaking valves show a similar effect as larger dead volumes. Thus, in all equations such as (407) (page 236) and (410), the volume lost by leaking needs to be added to the dead volume of the inlet V_e or outlet V_a valve, and both the flow rate and the pressure which can be generated by the pump are decreasing.

When a liquid is pumped and a *gas bubble is in the pump chamber*, the volume of the bubble is decreased and increased by the pressure change in the pump. This volume changes ΔV_B do not contribute to the flow rate of the pump and need to be subtracted from the volume V_F which is pumped with each pump cycle. The volume change of the bubble has a similar effect as the dead volumes of inlet V_e and outlet V_a valve (cf. Fig. 176 on page 233). When the pressure in the pump chamber is increased by the actuator, the dead volume of the inlet valve is displaced out of the pump and the volume of the bubble is decreased by ΔV_B, and when the pressure in the pump is decreased again, the dead volume of the outlet valve is flowing in backward direction and the volume of the bubble is increased before new liquid is taken into the pump. Therefore, the effect of a bubble is similar to the one of larger dead volumes of inlet and outlet valve.

The volume change ΔV_B of the bubble in the pump chamber can be calculated with the ideal gas law [(295) on page 167] which means that the product of pressure and volume of the bubble is constant (as long as the temperature does not change). When the inlet valve opens, the pressure and volume of the bubble are denoted by p_e and $V_{B,e}$, respectively. When the outlet vale opens and when there is a pressure of 101.3 kPa, they are denoted by p_a, $V_{B,a}$, and by p_0, $V_{B,0}$, respectively:

$$p_e \; V_{B,e} = p_a \; V_{B,a} = p_0 \; V_{B,0} \tag{413}$$

$$\Rightarrow \Delta V_B = V_{B,e} - V_{B,a} = \left(\frac{1}{p_e} - \frac{1}{p_a}\right) p_0 \; V_{B,0} = \frac{\Delta p_P}{p_e \; p_a} p_0 \; V_{B,0}. \tag{414}$$

Inserting this into (407) (page 236) as an additional dead volume of inlet and outlet valve yields:

$$\Phi_P = f_P \left\{ \left[\Delta V_{max} \left(1 - \frac{\Delta p_P}{\Delta p_{max}}\right) - V_e - \Delta V_B \right] \left[1 - e^{-\frac{\Delta p_{max}}{\Delta V_{max}} \frac{t_f}{R_{fl}}} \right] - V_a - \Delta V_B \right\} \tag{415}$$

As a consequence of the above equation, the pump is no longer able to pump any liquid when the bubble becomes too large. Thus, besides the problem that bubbles can block the valves because the gap over the valve seat is narrower than the feed channels (cf. page 118), the volume change of bubbles in the pump chamber decreases or even stops the flow. Special microstructures before a micropump can separate bubbles from the liquid flow to avoid such problems with bubbles (cf. page 118).

The volume of a bubble is limited by the dead volume V_P of the pump which is defined as the inner volume of the pump between the valve seats of inlet and outlet valve. The flow rate calculated with (415) can be larger than zero when the term in the curly braces is larger than zero. The exponential function can be assumed to be zero because the time of the delivering pulse of the pump will be made large to achieve a larger flow rate:

$$\Delta V_{max}\left(1-\frac{\Delta p_P}{\Delta p_{max}}\right) \geq V_e + V_a + 2\frac{\Delta p_P}{p_e\ p_a} p_0\ V_P \tag{416}$$

$$\Rightarrow V_P \leq \left[\Delta V_{max}\left(1-\frac{\Delta p_P}{\Delta p_{max}}\right) - V_e - V_a\right]\frac{p_e\ p_a}{2\ p_0\ \Delta p_P} \tag{417}$$

If the dead volume V_P of a pump is small enough to fulfill the above equation, the pump is also able to pump a gas. As a consequence, a pump with such a small dead volume is also self-priming, i.e., the pump can pump the air out of a pipe or hose connected to its inlet, and, this way, suck in a liquid. Besides this, it has to be concluded that the dead volume of a pump should be as small as possible to achieve a large flow rate and especially a large pressure.

According to (414), the volume change ΔV_B of a bubble in the pump chamber is proportional to the pressure difference over the pump Δp_P. That is, a bubble has only little effect on the flow rate when the pressure difference over the pump is small. The larger the counter pressure is, the more the flow rate is decreased by the volume change of the pump. In Fig. 185, this is shown for the case that the bubble is as large as the dead volume of the pump, i.e., only gas is pumped. The same parameters are assumed as for Fig. 182 and a radius R_Z of 65 μm. The characteristic curves of the pump pumping an incompressible fluid is shown together with the curves of a compressible gas and a dead volume V_P of the pump which is a factor of 2 (200 nL) and 4 (400 nL) larger than the maximum displacement ΔV_{max} of the actuator, respectively.

The maximum counter pressure $\Delta p_{P,m}$ which can be generated by a pump pumping a compressible fluid is found when the flow rate calculated with (415) is zero. This is the case when the term in the curly braces is zero. The exponential

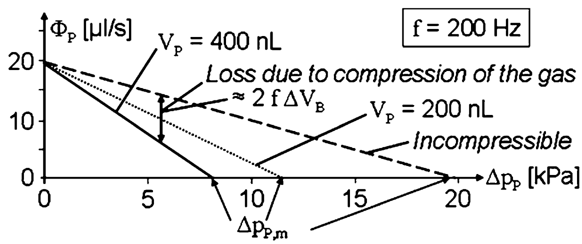


Fig. 185 Characteristic curve of a micropump pumping an incompressible fluid and a compressible gas with (theoretically) the same viscosity

function can be assumed to be zero because the time of the delivering pulse of the pump will be made large to achieve a larger counter pressure:

$$\Delta p_{P,m} = \frac{\Delta V_{max} - V_e - V_a}{2\frac{p_o}{p_e}\frac{V_p}{p_a} + \frac{\Delta V_{max}}{\Delta p_{max}}} \quad (418)$$

As shown above, both the maximum counter pressure and the maximum flow rate are reduced by the need to open or close the valves and to pump their dead volumes. This can be avoided when a *pump with active valves* is employed. As a result, the pump cycle is as shown in Fig. 174 instead of Fig. 176 (on page 230 and 233, respectively). The flow through the outlet valve is calculated with (401) and the flow passes through the outlet the quicker, the larger the radii of the feed channels and the valve are (cf. Fig. 175, page 232). Thus, a pump with active valves achieves a larger maximum flow rate and a larger maximum counter pressure. The reason for this is that the energy of the pump actuator is only used for pumping the fluid and not to open and close the valves. However, this energy is provided now by the actuators of the valves, and a similar result could be obtained with a somewhat larger pump actuator and passive valves.

A clear advantage of a pump with active valves is that the pump direction can be reversed by controlling valve switching. Therefore, a pump with active valves may replace two pumps and additional valves in certain applications. A special case of a pump with active valves is a peristaltic pump, which employs three or more equal pump chambers driven in consecutive order (cf. Fig. 186).

A pump delivering air which shows a dead volume of the valves as large as the pump chamber is shown in Fig. 187 [65]. In a pump chamber which is formed by two bell-shaped walls, there are mounted two membranes. In the membranes, there are orifices staggered to each other such that no air can pass through when the membranes are in touch to each other.

Therefore, the pump is sealed when both membranes are at the bottom of the pump chamber (Fig. 187a). Both membranes are insulated to the surrounding and contain electrodes. The membranes are held down by applying a voltage between the upper membrane and the bottom of the pump chamber. Then, voltage is applied between the

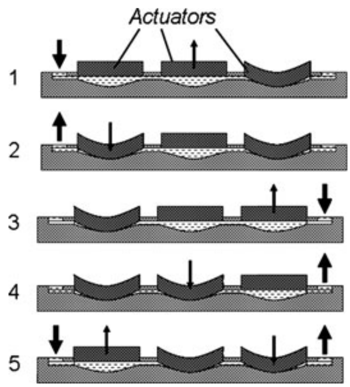


Fig. 186 Pump cycle of a peristaltic pump

lower membrane and the upper part of the pump chamber. This way, both membranes are moving up and delivering air to the outlet of the pump and taking in fresh air through the inlet (Fig. 187b). Since electrostatic forces are strong only at small distances, the bell-shaped walls facilitate moving of the membranes. The movement starts at the rim of the membranes where there is only a small distance to the upper wall and then continues until the entire membranes are in touch to the upper wall (Fig. 187c).

Then only the lower membrane is attracted down to the bottom of the pump chamber and no air is entering or leaving the pump, because the outlet is sealed by the upper membrane (Fig. 187d). During this membrane movement, the flow resistance of the orifices of the membrane needs to be overcome. The upper membrane follows the lower one in the next step and again no air enters or leaves the pump because the lower membrane seals the inlet (Fig. 187e).

Obviously, the air volume in the pump chamber in every pump cycle needs to pass through the membranes two times, and, therefore, is the dead volume of the "valves".

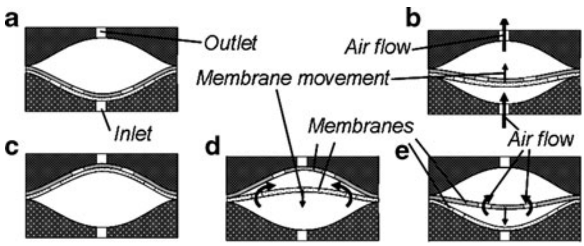


Fig. 187 Cycle of a pump with two electrostatically driven membranes [65]. © [1997] IEEE

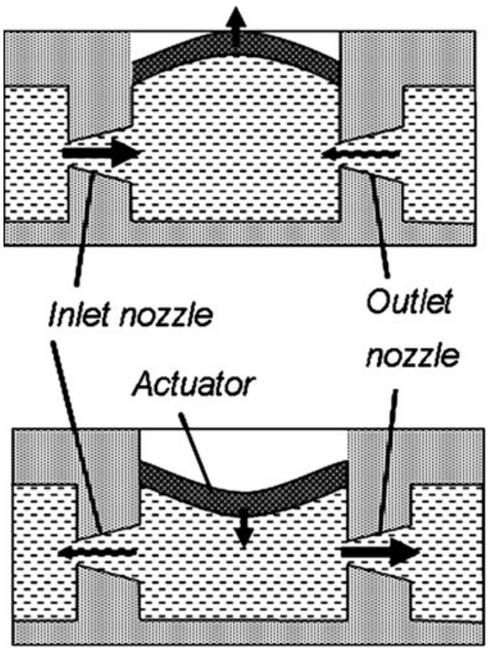


Fig. 188 Valveless micropump

For certain applications, it may be an advantage of this pump that it can be driven in backward direction.

It is also possible to design valves without any moving mechanical parts. Pumps employing such valves are sometimes called "valveless" pumps. Their valves consist of nozzles which show a larger flow resistance in one direction than in the opposite one (see Fig. 188). The back flow in the undesired direction is compensated by a larger flow in the desired one. However, such valves have a huge dead volume, and, as a consequence, they are not able to work against large counter pressures. Besides this, they are open in both directions when the actuator is not running and they achieve only a small efficiency. It is an advantage of such pumps that their valves are very rugged and easy to fabricate. However, other kinds of valves are also rugged enough.

The pumps described so far are all reciprocating pumps which generate a more or less pulsing flow. For some applications, this is not desirable. Solutions for that problem are *aperiodic pumps*. One aperiodic pumping principle has already been described earlier in this book: the electro-osmotic micropump (cf. page 122). Another possibility is to design a small chamber and fill it with a sorption agent which draws in a liquid continuously until it is completely filled [66].

Another interesting design is the electrohydrodynamic pump (EHD) [67]. An EHD consists of two gratings which spread across a channel. Between the gratings there is a certain distance. One grating is fabricated such that it shows a sharp edge pointing to the other grating. When the negative pole of a voltage is applied to the grating with the sharp edge and the positive to the other one, electrons are injected from the sharp edges into the liquid and they are accelerated towards the grating with the positive pole. Due to friction in the liquid not only the electrons but also the entire liquid is accelerated.

The EHD principle only works if the liquid is not conductive. Therefore, water cannot be delivered but certain oils. Large flows are achieved but only a very small pressure difference can be overcome.

Exercises

Problem 38

Figure E29 shows the cross-section of a micropump [68]. The properties of the pump are listed in the following table.

Diameter of pump chamber and actuator	12 mm
Deflection of the piezo actuator	20 μm
Length of the edge of the quadratic valves	1.5 mm
Length of the valve gap in the direction of the flow	500 μm
Stroke of the valve	30 μm
Diameter of the feed channels of the valve	0.4 mm
Length of the feed channels of each valve	1 mm
Viscosity of water	1 mPa s
Maximum pressure difference Δp_{max} generated by the actuator	100 kPa

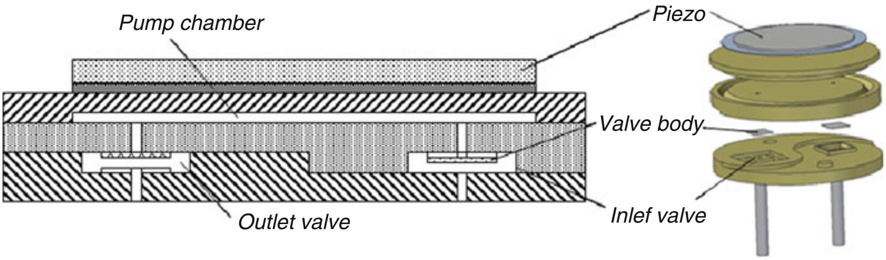
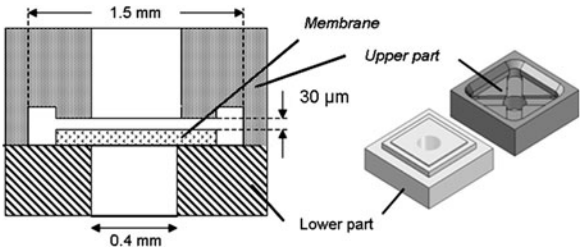


Fig. E29 Micropump with a piezo actuator and polymer plates as valve bodies. (Reprinted from [68] with permission from Elsevier)

Fig. E30 Valve of the micropump shown in Fig. E29. (Reprinted from [68] with permission from Elsevier)



(a) Please calculate the maximum volume change generated by the actuator.
(b) Please calculate the length of the pump pulse. At what frequency does the micropump achieve the maximum flow rate if the switching times of the valves are negligibly short and the maximum volume change generated by the actuator is 1 µL?
(c) In the valves of the pump, freely movable square plates from polymer are installed as valve bodies as shown in Fig. E30. The length of the edges of these plates is 1.4 mm. Please calculate the dead volume of the passive valves. Neglect the volume of the grooves in the upper part.
(d) What is the switching time of the passive valve according to (390) (page 223), if water is used as the fluid? Assume that the length of the feed channels and the pressure drop over them are 1 mm and 50 kPa, respectively.
(e) How could the design be improved?

Problem 39

It is your objective to design flap valves for a micropump. Assume that the mass of the fluid and the valve body can be neglected. As an approximation, assume that the pump is driven at its optimum frequency and that the stroke of the valves is large also at the beginning of the pump pulses. The following data are available for the calculation.

Maximum volume change generated by the actuator ΔV_{max}	600 nL
Maximum pressure difference generated by the actuator Δp_{max}	24 kPa
Pressure difference between inlet and outlet $p_a - p_e$	12 kPa
Length of the feed channels to the valves	400 µm
Viscosity of water	1 mPa s
Length of the valve beam	1 mm
Young's modulus of the beam	120 GPa
Thickness of the beam	10 µm

(a) Calculate the radius of feed channels and inlet and outlet (assumed to be equal) required for the maximum flow rate of the pump.
(b) What is the maximum flow rate that can be achieved?
(c) With which tolerance needs the radius of the valve orifice R_Z to be manufactured, if the flow rate may deviate from the maximum flow rate calculated in (b) not more than 0.1 mL/s? Assume for this calculation that all other dimensions are exactly kept to the desired measures. (Hint: For this problem, it is advantageous to employ a spreadsheet program.)

Problem 40

For this problem, the data of the following table are given:

Radius of the glass sphere	200 µm
Density of glass	2,600 kg/m^3
Radius of the feed channels	100 µm
Acceleration of gravity	9.81 m/s^2
Length of the feed channels	1 mm
Length of the valve gap	200 µm
Young's modulus of silicon	130 GPa
Width of the beam	600 µm
Thickness of the beam	20 µm
Viscosity of water	1 mPa s
Maximum pressure difference generated by the actuator Δp_{max}	200 kPa
Maximum volume change generated by the actuator ΔV_{max}	1 µL

Derive the equation for the flow of a micropump without counter pressure, if instead of a flap valve a freely moving sphere is employed in the passive valves. Please follow the subsequent questions:

(a) If the sphere is made of glass with a diameter of 400 µm, what is the minimum pressure difference required to lift the sphere in the gravitational field and to open the valve? Please neglect the buoyant force of the sphere in the fluid and any stiction.

(b) How large should be the stroke of the valve (Rule of thumb)?
(c) The stroke as a function of the pressure drop for a sphere cannot be found easily because it is a function of the flow situation inside of the valve. However, it appears to be reasonable to assume that the sphere is moved up to a stop 55 μm above the valve seat when the pressure difference is 60 Pa.
Please calculate how long a 600-μm wide and 20-μm thick beam from silicon needs to be which is employed instead of the sphere in (a). The beam shall show a stroke of 55 μm at a pressure difference of 60 Pa.
(d) What is the dead volume of the microvalve (stroke = 55 μm)?
(e) Calculate the switching time of the valve, if the effective pressure difference over the feed channels is 80 Pa.
(f) Please calculate the pump pulse time on the one hand with the approximation of widely opened valves and on the other hand with the effect of the valve gap.
(g) Why is it advantageous to limit the stroke of the valves (both for a sphere and a beam) by a stop above the valve seat?
(h) What is the maximum pump frequency, if the switching time of the valves is taken into account and the pump pulse time is 150 μs? (The question whether the actuator is able to generate this frequency is not discussed here.)
(i) What is the delivered flow volume per pump pulse without counter pressure?
(j) What is the maximum flow rate of the pump without counter pressure?

Problem 41

For a diagnostic device, a liquid flow (η = 1 mPa s) of 1.5 μL/s shall be delivered against a pressure difference of 50 kPa. You have micropumps with flap valves on stock which fulfill the following specifications:

Radius of the feed channels	24 μm
Length of the feed channels	0.5 mm
Length of the valve gap	200 μm
Young's modulus of the beam	0.6 GPa
Length of the beam	1 mm
Thickness of the beam	15 μm
Maximum pressure difference generated by the actuator Δp_{max}	100 kPa
Maximum volume change generated by the actuator ΔV_{max}	50 μL

(a) Can the objective be fulfilled by the micropump?
(b) The flow rate at a counter pressure of 50 kPa now shall be enhanced. To achieve this, you first try to change the size of the feed channels. What radius should the feed channels obtain to achieve the maximum possible flow rate at the given counter pressure? How large is the maximum flow rate? How much can the flow rate be enhanced by changing the diameter of the feed channels? (A spreadsheet program or a programmable pocket calculator will help to solve this problem.)

(c) Now prove the alternative that two down sized micropumps driven in series are employed, if each of them achieves half of the flow rate calculated in (a) but the same maximum pressure. What is in this case the flow rate at 50 kPa counter pressure?
(d) Calculate the fluidic power of the pump in (b) at a counter pressure of 50 kPa.

Problem 42

For the following problem, the data of the following table are given:

Diameter of the pump chamber	4 mm
Minimum flow rate	100 µL/h
Maximum counter pressure	20 kPa
Piezo electric modulus d_{31}	-2.14×10^{-10} m/V
Thickness of the piezo layer	100 µm
Absolute permittivity	8.9×10^{-12} A s/(V m)
Relative permittivity of air	1
Young's modulus of piezo ceramic	66.6 GPa
Young's modulus of silicon dioxide	73 GPa
Gas constant R_G	8.314 J/(mol K)

For an insulin pump, an actuator needs to be chosen. An insulin pump can be mounted outside of the skin and deliver through a catheter into the body. As an alternative, the pump can be implanted into the body. Figure E31 shows an example of an implantable micropump.

(a) Please calculate the voltage required to achieve the pressure difference with the piezo actuator. The piezo has the same diameter as the pump chamber and is glued onto a carrier from silicon dioxide, 50 µm in thickness. The thickness of the glue can be neglected.
(b) The voltage calculated in (a) is just enough to overcome the counter pressure but is not sufficient to deliver against this pressure. Therefore, you decide to apply 128 V. Please calculate the frequency at which the pump needs to run to achieve the specified flow rate at a counter pressure of 20 kPa. Do not take into account the dead volume of the valves.

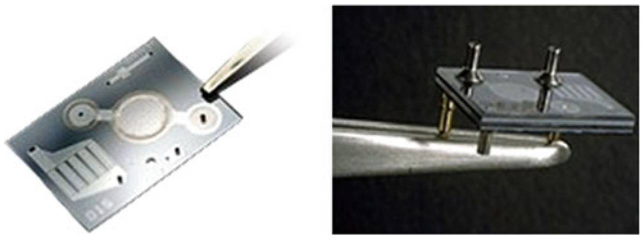


Fig. E31 Implantable micropump [69] (Courtesy of Debiotech SA/Switzerland)

(c) Please calculate the minimum voltage required to achieve the specified pressure difference by an electrostatic actuator. Neglect the elastic force of the membrane and assume that there is air between the electrodes. The distance between membrane and counter electrode is 10 μm.
(d) If for an implantable pump for safety reasons there is allowed only a voltage smaller than 36 V, how needs the design to be changed to achieve the required counter pressure.
(e) How much would a thermo-pneumatic actuator need to be heated up to achieve the required pressure difference?
(f) For some applications, a micropump driven hydraulically by the blood pressure could be of interest. It could, e.g., be self-controlling when some drug against high blood pressure is delivered more when the blood pressure or the pulse beat is high. How large would the flow rate of such a pump be if its displaced volume per pulse is 0.02 μL at a typical heart frequency of 1 Hz?