Modelling and experimental characterisation of a magnetic shuttle pump for microfluidic applications

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Microfluidic technology witnessed a fast growth in recent years thanks to its diverse nature that allows its use in a wide range of industries including microelectronics, aerospace, telecommunications, biomedical and pharmaceutical. One of the limiting issues for the implementation of microfluidics in high end electronics or biomedical devices is that pumps are not able to develop the required flow rates and pressures.

A novel magnetic shuttle pump (MSP) technology that can achieve class-leading pressure and flow rate and a numerical model is presented in this paper. The MSP technology consists of an oscillating neodymium ring shuttle magnet housed in a solenoid driver. Two counter-wound copper coils are used to oscillate the shuttle magnet.

The numerical model couples the electromagnetic and fluidic properties of the MSP by taking into account the forces acting on the shuttle magnet. The model is used to predict the pump characteristics of two MSPs with different size: the MSP1.7 with overall volume 1.7 cm³ and MSP3.3 with overall volume 3.3 cm³. Simulations and experimental characterisation were carried out considering an electric driving power of 1W. Experimentally, a maximum pressure \( P_{\text{max}} = 43.53 \text{kPa} \) and a maximum flow rate \( Q = 46.69 \text{ml/min} \) were achieved by the MSP1.7, while a maximum pressure \( P_{\text{max}} = 21.74 \text{kPa} \) and a maximum flow rate \( Q = 205.99 \text{ml/min} \) were achieved by the MSP3.3. Due to the close agreement between the experimental and simulated data, the model can be used in the future to modify the design of the MSP to achieve the required Pressure/Flow characteristics.

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1. Introduction

Micropumping has emerged as a critical research area for many electronic and biological applications. Since a small foot-print (~1–4 cm³) is often required, commonly used rotary pumps are unsuitable due to their larger size, pumping power and high power consumptions. Based on these issues novel pumping methods have been developed and miniaturised to be used in the micro-scale. These so called μpumps are used in drug delivery systems [1–3], lab-on-chip [4–6], biochemistry [7], controlled fuel delivery in engines [8], fuel cells [9] and micro mixing [10].

Several researchers focused on the development of μpumps for single- or two-phase cooling of electronic components [11–13]. However, one main issue is that flow rates of over several hundreds of millilitres per minute (ml/min) are required for single-phase chip cooling [11]. In addition to obtaining the high flow rates, the pump needs to overcome high pressures (~10–100 kPa) to force the fluid through the microchannels. Additional challenges include cost, power consumption, volumetric footprint and fabrication methods [14].

Several different kind of μpumps have been developed and can be divided into two major categories [15]: displacement μpumps, which use moving boundaries to impress pressure forces on the fluid; and dynamic μpumps, which increase the momentum and pressure of the working fluid by continuously adding energy to the fluid. Most commonly reported μpumps in literature are displacement pumps which use a diaphragm or a membrane as the moving part. Two different methods are generally used to displace the diaphragm: piezoelectric actuation or electromagnetic actuation.

In the first case, the diaphragm is generally made of a piezoelectric material and when a sinusoidal voltage is applied, the diaphragm oscillates. One of the main drawbacks of piezoelectric actuated μpump is that large voltages are needed to actuate the piezoelectric membrane, as in [16–18]. The devices presented in [17] and [18] can achieve flow rates of 220 ml/min and 200 ml/min,
respectively, but the size of the two devices (79.4 cm³ and 43.2 cm³) are not suitable for cooling applications. Smaller pumps were fabricated but they do not develop the flow rates required for chip cooling, as in [19].

In electromagnetic actuated μpumps, a permanent magnet is generally attached to a membrane and a coil is used to generate a magnetic field that deflects the membrane, similar to what is done in [20,23]. Said et al. [21] proposed, instead, to use membrane made of a magnetic polymer and an attached permanent magnet. Even though lower voltages are necessary to actuate this type of pumps, flow rates and pressures are not in the desired range for cooling applications.

To overcome these problems, a novel displacement pump based on a patent pending [22] counter-wound solenoid technology is presented in this paper. The magnetic shuttle pump (MSP) features two counter-wound solenoid coils, which are used to oscillate a neodymium ring shuttle magnet. A valve is present inside the shuttle magnet to induce a net fluidic flow. Two different pumps, MSP1.7 and MSP3.3, are considered in this paper with overall volume 1.7 cm³ and 3.3 cm³, respectively.

A numerical model is also presented in the paper and simulations are carried out to verify its capability of predicting the two pumps characteristics. Finite element analysis is used to determine the loss coefficients of the valves, while the balance of the electromagnetic forces acting on the shuttle magnet is considered to couple the shuttle dynamics with the fluidic properties of the pump.

Despite the small volume (1.7 cm³), the MSP1.7, can achieve a backpressure of 43.53 kPa and a maximum flow rate of 46.69 ml/min at a power consumption of only 1 W. The MSP3.3 instead can achieve a backpressure of 21.74 kPa and maximum flow rate of 205.99 ml/min at a power consumption of 1 W.

In the following sections the numerical model will be derived and simulated and experimental pump characteristics will be presented as a validation of the model. The two MSPs will also be compared with commercial off-the-shelf pumps to show the benefit of the proposed approach.

2. System of interests

The magnetic shuttle pumps (MSPs) MSP1.7 and MSP3.3 examined in this paper are shown in Fig. 1. The diameter of the MSP1.7 is 10.5 mm and the length is 29.5 mm for an overall volume of 1.7 cm³, while the diameter of the MSP3.3 is 14.1 mm and the length is 34 mm for an overall volume of 3.3 cm³.

A section of the pumps is shown in Fig. 2. The pump exploits a novel counter-wound solenoid coil technology to oscillate a shuttle-style neodymium ring magnet as illustrated in Fig. 2. A ceramic sphere and a titanium spacer are present inside the shuttle magnet, while two ceramic spheres and a stainless steel spacer are present in the input connector to realise one way high frequency valves which induce the flow. The valve in the shuttle magnet will be referred as shuttle valve, while the valve inside the input connector will be referred as input valve. Two further neodymium ring magnets are placed inside the input and output connectors and act as magnetic springs to hold the shuttle in the centre of the driver, where the intensity of the magnetic force due to the coils is larger.

The neodymium magnets are coated in Nickel while the ceramic spheres are made of Silicon Nitride. Both magnets and balls are off-the-shelf components. The other parts of the MSP1.7 and MSP3.3 are made in Stainless Steel 316 and were CNC machined in house.

Fig. 3 shows the shuttle position and the valves functioning when a sinusoidal voltage is applied to the counter-wound coils. At zero input voltage, the shuttle is held in the centre of the driver by the two magnetic springs. During the positive half of the driving cycle, the counter wound coils generate two magnetic fields with south poles at the central position and the shuttle is moved towards the right. As a consequence of movement, the input valve opens letting the fluid enter the left chamber of the pump and the shuttle valve closes letting the fluid leave the right chamber. During the negative half of the driving cycle, the coils generate magnetic fields with the north poles in the centre and the shuttle is moved towards the left. As a consequence, the input valve closes not allowing the fluid to enter the left chamber and the shuttle valve opens allowing the fluid that is already in the left chamber to move to the right chamber. Due to the oscillation of the magnet and the presence of the two internal valves, a net fluid flow is produced.

The characterisation of the pump was performed using the setup in Fig. 4. The pressure developed by the pump is measured by an Omega PX409 pressure transducer while the flow rate produced is measured by a Bronkhorst-L30 flow meter. In both cases, the data are acquired by a NI PXIe-6363 data acquisition (DAQ) card. The sinusoidal voltage, used to drive the pump, is generated by the NI PXIe-6363 and amplified by a Bruel & Kjaer 704 power amplifier. The electrical input power delivered to the pump is measured using a Voltech PM1200 power analyser and recorded with the NI DAQ card. A secondary gear pump (Cole-Parmer) is required to overcome the fluidic resistance due to the tubing, the connectors and, especially, the flow meter, in order to characterise the pumping.
at low pressure. Finally, an Omega SFV12 flow restrictor is placed in the system to vary the fluidic resistance load on the MSP. Both restrictor and secondary pump are controlled by the NI PXIe-6363.

3. Modelling of the magnetic shuttle pump

In the next section a numerical model for the MSP will be derived by coupling the electromagnetic and fluidic properties of the pump. In the first subsection, the electromagnetic force generated by the two coils and the magnetic force due to the magnetic springs will be considered in order to calculate the maximum pressure and the maximum flow rate developed by the MSP. In the latter subsection, COMSOL Multiphysics will be used to derive the loss coefficients of the two valves.

3.1. Modelling of the electromagnetic forces

The principle of operation of the MSP described in the previous section is based on the Biot–Savart equation that expresses the magnetic field generated by an electric current: when an electric current flows in the coils, a magnetic field is generated and the shuttle magnet experiences a force, moving according to Fig. 3.

A numerical model, that couples the electromagnetic and fluidic properties of the MSP was developed. The model takes into account the balance of the electromagnetic force generated by the two coils and the force due to the magnetic springs in order to calculate the displacement of the shuttle magnet.

To couple the dynamics with the fluidic properties, however, few assumptions were made:

1. the input valve and the shuttle valve are ideal, i.e. no leakage was considered;
2. during the positive half of the driving cycle, the shuttle valve is closed and the shuttle magnet acts as a piston allowing fluid to leave the pump;
3. during the negative half of the cycle, the shuttle valve is open, the input valve is closed and the overall flow rate at the outlet is zero;
4. the fluid cannot flow between the shuttle magnet and the wall of the internal cavity;
5. the fluid is incompressible.

Fig. 5 can be used to model the coils-magnets arrangement. Due to the symmetry respect to the axis of the pump, only one half is represented in the picture.
Four different forces act on the shuttle magnet: the electromagnetic force due to the two coils ($F_{\text{coil}1}$ and $F_{\text{coil}2}$), the magnetic force due to the two small ring magnets ($F_{\text{spring}1}$ and $F_{\text{spring}2}$) and the fluidic resistance ($F_{\text{res}}$) due to the fact that the shuttle is moving into a liquid. The equation of motion is reported in Eq. (1) and it will be used to derive the maximum pressure and maximum flow rate achievable by the MSP1.7 and MSP3.3 at an input electrical power equal to 1 W.

$$M^2 = F_{\text{spring}1} - F_{\text{spring}2} + F_{\text{coil}1} - F_{\text{coil}2} - F_{\text{res}}$$  \hspace{1cm} (1)

where $M$ is the mass of the shuttle.

According to [23], a permanent magnet and a single-layer coil are electromagnetic equivalent since a surface current density can be associated to the magnet and the same expression for coil-magnet forces or magnet-magnet forces can be used. Therefore, starting from the force between two disk magnets, it is possible to derive the force between two ring magnets (hence, $F_{\text{spring}1}$ and $F_{\text{spring}2}$) and the force between the shuttle ring magnet and the multi-layer coils (hence, $F_{\text{coil}1}$ and $F_{\text{coil}2}$).

The force between two permanent disk magnets ($F_{\text{disk}}$) of strength $B$: is given by [24]:

$$F_{\text{disk}}(R_{m1}, R_{m2}, l_{m1}, l_{m2}, z) = \frac{J_1 J_2}{2\mu_0} \sum_{n_1, n_2} e_1 e_2 m_1 m_2 m_3 f_s$$  \hspace{1cm} (2)

where $J_1 = J_2 = B$: is the strength of the permanent magnets, $R_{m1}$ and $R_{m2}$ are the radii of the disk magnets, $l_{m1}$ and $l_{m2}$ are the lengths of the disk magnets, $z$ is the distance between the centres of mass, and $f_s$ is given by:

$$f_s = K(m_4) - \frac{1}{m_2} E(m_4) + \left[ \frac{m_2^2}{m_3^2} - 1 \right] \Pi \left( \frac{m_4}{1 - m_2/m_4} \right)$$  \hspace{1cm} (3)

with parameters:

$$m_1 = z - \frac{1}{2} e_1 l_{m1} - \frac{1}{2} e_2 l_{m2}, \quad m_2 = \frac{[R_{m1} - R_{m2}]^2}{m_3^2} + 1, \quad m_3 = \sqrt{[R_{m2} + R_{m1}^2 + m_1^2], \quad m_4 = \frac{4 R_{m2} R_{m1}}{m_3}.}$$  \hspace{1cm} (4)

The functions $K(m)$, $E(m)$ and $\Pi(n|m)$ are the complete first, second and third elliptic integrals with parameter $m$.

Eq. (2) can be used to determine the force between two permanent ring magnets of internal radius $r_{m1,m2}$, external radius $R_{m1,m2}$, length $l_{m1,m2}$ and surface current density $J_{1,2} = B$:.

The flow resistance for two ring magnets is given by:

$$F_{\text{ring}}(r_{m1,m2}, R_{m1,m2}, l_{m1,m2}, z)$$ between two ring magnets is given by:

$$F_{\text{ring}} = F_{\text{disk}}(R_{m1,m2}, l_{m1}, l_{m2}, z)$$

$$+ F_{\text{disk}}(r_{m1,m2}, l_{m1}, l_{m2}, z)$$

$$- F_{\text{disk}}(r_{m1,m2}, l_{m1}, l_{m2}, z)$$

$$- F_{\text{disk}}(R_{m1,m2}, l_{m1}, l_{m2}, z)$$  \hspace{1cm} (5)

With reference to Fig. 5, the force $F_{\text{ring}}$ represents also the force due to the magnetic springs:

$$F_{\text{spring}1} = F_{\text{ring}}(r_{m1,m2}, R_{m1,m2}, l_{m1}, l_{m2}, z)$$

$$F_{\text{spring}2} = F_{\text{ring}}(r_{m1,m2}, R_{m1,m2}, l_{m1}, l_{m2}, z)$$  \hspace{1cm} (6)

The force between one of the two multi-layer coils and the shuttle ring magnet is modelled using the 'shell method' [23]; each radial layer of turns $n$ is represented as a single-layer coil of radius:

$$r(n) = r_e + \frac{n_r - 1}{N_r - 1} [R_e - r_e]$$  \hspace{1cm} (7)

where $N_r$ is the total number of radial layers (as shown in Fig. 5).

Each single-layer coil is equivalent to a ring magnet of internal radius $r_{lm1} = 0$, external radius $R_{lm} = r$, length $l_m = l_c$ and current density $J_{lm} = \mu_0 N_r l_c l_m$, where $I$ is the current in the coil. Therefore the force between each single-layer coil and the shuttle magnet is given by Eq. (5):

$$F_s(0, r_m, r, R_m, l_c, l_m, z) = F_{\text{ring}}(0, r_m, r, R_m, l_c, l_m, z)$$  \hspace{1cm} (8)

According to [23], the total force between the single-layer coils and the shuttle magnet ($F_{\text{coil}1}$ and $F_{\text{coil}2}$) is calculated by superposition of the forces between each single-layer coil and the magnet, considering current densities $J_{coil1} = \mu_0 N_r l_c l_m$ and $J_{coil2} = -\mu_0 N_r l_c l_m$:

$$F_{\text{coil}1} = \frac{1}{N_r} \sum_{n_r=1}^{N_r} F_s(0, r_m, r, R_m, l_c, l_m, z_1),$$

$$F_{\text{coil}2} = \frac{1}{N_r} \sum_{n_r=1}^{N_r} F_s(0, r_m, r, R_m, l_c, l_m, z_2).$$  \hspace{1cm} (9)

To determine the fluidic resistance, one full driving cycle must be considered. During the positive half, the shuttle valve is closed, while the input valve is open: the shuttle magnet experience a force due to the pressure drop at the input. During the negative half, the shuttle valve is open, while the input valve is closed: the shuttle magnet experience, therefore, a force due to the pressure drop at the entrance of the shuttle magnet. The fluidic resistance can be expressed according to the following equations:

$$F_{\text{res}}(z) = \begin{cases} k_{\text{input}} A_{\text{shuttle}}^2 z^2, & \text{if } z > 0 \\ k_{\text{shuttle}} A_{\text{shuttle}}^2 z^2, & \text{if } z < 0 \end{cases}$$  \hspace{1cm} (10)

where $k_{\text{input}}$ and $k_{\text{shuttle}}$ are the loss coefficients of the input valve and of the shuttle valve, respectively, and $A_{\text{shuttle}}$ is the surface area of the shuttle magnet.

Loss coefficients for the input valve and for the shuttle valve will be derived in Section 3.2 by using Comsol Multiphysics.

The dynamic of the shuttle magnet can be obtained by solving numerically Eqs. (1), (6), (9) and (10) and it can be used to calculate the maximum pressure and the maximum flow rate.

The maximum pressure is achieved when the outlet is blocked and, since the fluid is incompressible, the shuttle magnet cannot move. Therefore, the maximum pressure is achieved when the shuttle is in middle position and it is given by:

$$P_{\text{max}} = \frac{1}{T} \int_0^T \left( F_{\text{coil}1} l_2 - l_1 / 2 + F_{\text{coil}2} l_2 - l_1 / 2 \right)$$  \hspace{1cm} (11)
where \( T \) is the period of the electrical current in the coils.

The maximum flow rate \( (\dot{Q}) \) instead is given by the fluid displaced by the shuttle magnet:

\[
\dot{Q}_{\text{max}} = \frac{1}{T} \int_0^T v A 
\]

(12)

where \( v \) is the velocity of the shuttle as derived from the integration of Eq. (1). \( T \) is the period of the current in the coils and \( A \) is the area of the cavity where the shuttle magnet oscillates.

A linear pump characteristic was considered and the intermediate pressure/flow rate conditions are determined as:

\[
P = -\frac{P_{\text{max}}}{\dot{Q}_{\text{max}}} \dot{Q} + P_{\text{max}}
\]

(13)

The input valve and the shuttle valve were modelled in COMSOL Multiphysics to calculate \( k_{\text{input}} \) and \( k_{\text{shuttle}} \) for both the MSP1.7 and MSP3.3. Simulated data and experimental results for the loss coefficients are presented in the following section (Section 3.2). In Section 4, Eq. (1) will be solved numerically with Runge-Kutta algorithm using the parameters reported in Table 1 to calculate \( P_{\text{max}} \) and \( Q_{\text{max}} \) at the electrical input power of 1 W and the simulated results are compared with the experimental data.

### Table 1

| Geometrical and coil parameters used as input to the model. | \( l_m \) (mm) | \( r_m \) (mm) | \( R_m \) (mm) | \( l_c \) (mm) | \( r_c \) (mm) | \( N_l \) | \( N_z \) |
|----------------|----------------|----------------|----------------|----------------|----------------|--------|--------|
| MSP1.7         | 12             | 0.75           | 2              | 8.5            | 2.1            | 11     | 33     |
| MSP3.3         | 11.5           | 2.5            | 3.8            | 8.5            | 3.9            | 11     | 33     |

3.2. Modelling of the loss coefficients

The input valve and the shuttle valve for the MSP1.7 and the MSP3.3 were modelled in COMSOL Multiphysics to calculate the loss coefficients \( k_{\text{input}} \) and \( k_{\text{shuttle}} \) that are used in Eq. (10) to derive the fluidic resistances.

To evaluate the type of flow present in the system and hence to decide the most appropriate physics to use in COMSOL, the Reynolds number was calculated according to:

\[
Re = \frac{\rho u D}{\mu} = \frac{4 \rho \dot{Q}}{\pi \mu D}
\]

(14)

where:

1. \( \rho = 10^3 \) kg m is the density of water;
2. \( \mu = 10^{-3} \) Pa s is the dynamic viscosity of water at 25 °C;
3. \( D = 4 \) mm is the diameter of the pipe;
4. \( u = Q/A \) is the mean velocity of the fluid;
5. \( Q \) is the fluid flow rate;
6. \( A = \pi(D/2)^2 \) is the area of the pipe.

It is expected that the MSP1.7 and MSP3.3 develop flow rates below 300 ml/min that, by using Eq. (14), would give Reynolds numbers lower than \( Re_{max} = 1591 \). The MSPs develop, hence, laminar flow regimes and the “Laminar Flow” physics was used in COMSOL. To simplify the geometry, only the internal part of the input valve and of the shuttle valve were considered and the geometry used for the two valves is shown in Fig. 6a and b for the MSP1.7. Similar geometries were used for the MSP3.3. Normal inflow velocity \( u = Q/A, \) \( Q \) is the flow rate, \( A \) is the pipe area) was applied at the inlet boundary and it was assumed that the pressure at the outlet boundary was 0. The pressure drop \( P \) across the valve was calculated as the difference between the average pressure on the inlet boundary and the average pressure on the outlet boundary (that is 0).

Figs. 7a and b show the velocity field lines at \( Q = 50 \) ml min\(^{-1} \) for the MSP1.7 input valve and for the MSP1.7 shuttle valve, respectively. As expected for a laminar flow, field lines are parallel.

According to [23] \( (P) \) across a valve as function of the flow rate \( (Q) \) is given by:

\[
P = \frac{\rho}{2} \xi \left( \frac{Q}{A} \right)^2 = k_v Q^2
\]

where \( \rho \) is the density of the fluid, \( \xi \) is a pressure-loss coefficient and \( A \) is the area of the throat of the valve. The coefficient \( k_v \) can be obtained by fitting the trend of \( P \) as a function of \( Q \) with a polynomial of order 2. Simulation were, hence, carried out considering a range of flow rate at the inlet boundary to measure the trend of \( P \) as function of \( Q, \) Fig. 8a and b show characteristics of the MSP1.7 input valve and shuttle valve and their fit. Similar plots were obtained for the MSP3.3.

The loss coefficients for the two valves for the MSP1.7 and MSP3.3 are reported in Table 2. These values are used in Eq. (10) to derive the force due to the fluidic resistance and will be verified experimentally in Section 4.1.

The loss coefficients for the two valves for the MSP1.7 and MSP3.3 are reported in Table 2. These values are used in Eq. (10) to derive the force due to the fluidic resistance and will be verified experimentally in Section 4.1.

### 4. Results

The MSPs described in the previous sections were fabricated, modelled and tested using the experimental setup reported in Fig. 4. Geometrical and coil parameters of the pumps are reported in Table 1 and are used as input parameters for the model described in the previous section. The first subsection presents the experimental characterisation of the two pumps at the electrical input power of 1 W. Loss coefficients for the valves will also be presented to validate the COMSOL Multiphysics model presented in Section 3.2. The second subsection instead will present the simulated results.

#### 4.1. Experimental characterisation

The valves for the MSP1.7 and for the MSP3.3 were characterised in open condition in order to measure the loss coefficient \( k_v. \) As seen in Section 3.2, the coefficient \( k_v \) can be obtained by fitting the trend of \( P \) as function of \( Q \) with a second order polynomial (Eq. (15)). Similar plots were obtained for the MSP1.7 shuttle valve and for the two MSP3.3 valves. Three independent tests were carried out to measure the loss coefficients and their values are reported in Table 3, with associated mean values and errors. Fig. 9 show, the characteristic of the input valve for the MSP1.7 and its fit for one of the three tests carried out.

### Table 2
Loss coefficients for input and shuttle valves for MSP1.7 and MSP3.3

| Valve Type | \( k_v^{input} \times 10^{15} \) | \( k_v^{shuttle} \times 10^{15} \) |
|-----------|-------------------------------|----------------------------------|
| MSP1.7    | 5.884 \times 10^{15} kgm\(^{-7}\) | 3.497 \times 10^{15} kgm\(^{-7}\) |
| MSP3.3    | 2.260 \times 10^{14} kgm\(^{-7}\) | 1.154 \times 10^{14} kgm\(^{-7}\) |

### Table 3
Experimental loss coefficients for the MSP1.7 and the MSP3.3. Units are in kgm\(^{-7}\).

| Valve Type | \( k_v^{input} \times 10^{15} \) | \( k_v^{shuttle} \times 10^{15} \) | \( \Delta k_v^{input} \times 10^{15} \) | \( \Delta k_v^{shuttle} \times 10^{15} \) |
|-----------|-------------------------------|----------------------------------|---------------------------------|---------------------------------|
| MSP1.7    | 5.940                          | 3.165                            | 0.029                           | 3.199                           | 0.028                           |
| Test 1    | 5.981                          | 3.210                            | 0.022                           | 3.221                           |                                  |
| Test 2    | 5.999                          |                                  |                                  |                                  |                                  |
| Test 3    | 2.345                          | 1.114                            | 0.023                           | 1.069                           | 0.023                           |
| MSP3.3    | 2.299                          | 1.073                            |                                  |                                  |                                  |
| Test 1    | 2.324                          | 1.085                            | 0.023                           |                                  |                                  |
| Test 2    | 2.320                          |                                  |                                  |                                  |                                  |
| Test 3    | 2.329                          |                                  |                                  |                                  |                                  |
Fig. 9. MSP1.7 experimental input valve characteristics for only one of the three test carried out.

![Graph showing input valve characteristics for MSP1.7](image)

Fig. 10. Experimental characteristics of the MSP1.7 and for the MSP3.3 at \( W_{el} = 1 \) W for only one of the three test carried out.

![Graph showing experiment characteristics for MSP1.7 and MSP3.3](image)

The loss coefficient of the MSP1.7 input valve was measured as \( k_{\text{input}} = 5.973 \times 10^{15} \pm 0.029 \times 10^{15} \text{kglm}^{-2} \), while the loss coefficient for the MSP1.7 shuttle valve was \( k_{\text{shuttle}} = 3.199 \times 10^{15} \pm 0.028 \times 10^{15} \text{kglm}^{-2} \). The loss coefficient of the MSP3.3 input valve was measured as \( k_{\text{input}} = 2.324 \times 10^{14} \pm 0.023 \times 10^{14} \text{kglm}^{-2} \), while the loss coefficient for the MSP3.3 shuttle valve was \( k_{\text{shuttle}} = 1.085 \times 10^{14} \pm 0.023 \times 10^{14} \text{kglm}^{-2} \).

The pumps were then characterised using the setup in Fig. 4 using a sinusoidal voltage of frequency \( f = 40 \) Hz at the electric power \( W_{el} = 1 \) W. Fig. 10 shows the pump characteristics for the MSP1.7 and for the MSP3.3. For clarity, only one of the three datasets is illustrated in the plots. For each data point, equilibrium was reached and the pressure and flow rate were averaged over a period of 7 s.

The values of the maximum pressures \( p_{\text{max}} \) and maximum flow rate \( Q_{\text{max}} \) for \( W_{el} = 1 \) W for the two pumps are reported in Table 4, respectively, with associated mean values and errors.

Fig. 11 shows instead the efficiency \( \eta \) of the two pumps at \( W_{el} \), calculated as:

\[
\eta = \frac{P_{\text{max}} Q_{\text{max}}}{W_{el}}.
\]

The MSP1.7 achieves at 1 W a maximum pressure and flow rate of \( P_{\text{max}} = 43.53 \pm 0.24 \text{kPa} \) and \( Q_{\text{max}} = 46.69 \pm 0.32 \text{ml/min} \), corresponding to a pumping power \( W_{\text{pumping}} = P_{\text{max}} Q_{\text{max}} = 2032.42 \pm 25.14 \text{kPa ml/min} \) and a pumping power per volume equal to \( 1191.63 \pm 16.41 \text{kPa ml min}^{-1}/\text{cm}^2 \), where the volume of the pump is 1.705 \pm 0.002 cm\(^3\) (error on each pump dimensions is 0.01 mm).

The MSP 3.3 achieves, instead, at 1 W a maximum pressure and flow rate of \( P_{\text{max}} = 21.74 \pm 0.13 \text{kPa} \) and \( Q_{\text{max}} = 205.99 \pm 0.18 \text{ml/min} \), corresponding to a pumping power \( W_{\text{pumping}} = P_{\text{max}} Q_{\text{max}} = 4478.22 \pm 30.69 \text{kPa ml/min} \) and a pumping power per volume equal to \( 1346.83 \pm 10.65 \text{kPa ml min}^{-1}/\text{cm}^2 \), where the volume of the pump is 3.325 \pm 0.003 cm\(^3\) (error on each pump dimension is 0.01 mm).

The maximum efficiency \( \eta = 1.097 \pm 0.075 \% \) is achieved by the MSP3.3 for a flow rate of 101.06 \pm 0.09 ml/min. As illustrated in Fig. 11, the optimal range of operation for the MSP1.7 is between 10 ml/min and 30 ml/min, while for the MSP3.3 is between 20 ml/min and 180 ml/min.

4.2. Numerical results

Eqs. (1), (11), (12) and (13) were solved numerically using Runge-Kutta order four algorithm. The parameters reported in Table 1 and the \( k_i \) values calculated by using COMSOL were used. The remanent magnetic flux density of the magnets was considered equal to \( B_r = 1.4 \text{T} \). The same sinusoidal voltage signal and the same electrical power (\( W_{el} = 1 \) W) of the previous subsection are considered.

Fig. 11 shows the predicted pump characteristics for the two pumps, while the predicted efficiencies are reported in Fig. 13. The experimental data presented in Section 4.1 are also presented for better comparison in both Figs. 12 and 13.
The maximum simulated pressure and simulated flow rate achieved by the MSP1.7 at 1 W are $P_{\text{max}} = 42.11$ kPa and $Q_{\text{max}} = 48.00$ ml/min, corresponding to a pumping power $W_{\text{pumping}} = P_{\text{max}} Q_{\text{max}} = 2021.28$ kPa ml/min and a pumping power per volume equal to 1188.99 kPa ml min$^{-1}$/cm$^3$, where the volume of the pump is 1.7 cm$^3$.

The maximum simulated pressure and flow rate that the MSP3.3 achieves, instead, are $P_{\text{max}} = 23.05$ kPa and $Q_{\text{max}} = 226.00$ ml/min, corresponding to a pumping power $W_{\text{pumping}} = P_{\text{max}} Q_{\text{max}} = 5209.3$ kPa ml/min and a pumping power per volume equal to 1578.58 kPa ml min$^{-1}$/cm$^3$, where the volume of the pump is 3.3 cm$^3$.

### Table 5

|        | $P_{\text{max}}$ kPa | $P_{\text{pmax}}$ kPa | $\delta P$ | $Q_{\text{max}}$ ml/min | $Q_{\text{pmax}}$ ml/min | $\delta Q$ |
|--------|-----------------------|------------------------|------------|--------------------------|--------------------------|------------|
| MSP1.7 | 43.53                 | 42.11                  | -1.42      | 46.69                    | 48.00                    | 1.31       |
| MSP3.3 | 21.74                 | 23.05                  | 1.31       | 205.99                   | 226.00                   | 20.01      |

The maximum simulated efficiency achieved by the MSP1.7 is $\eta = 0.85\%$ for a flow rate of 24 ml/min, while the maximum simulated efficiency achieved by the MSP3.3 is $\eta = 2.17\%$ for a flow rate of 114 ml/min. Similar to what was found experimentally, the predicted optimal range of operation of the MSP1.7 is between 10 ml/min and 40 ml/min while the predicted optimal range of operation of the MSP3.3 is between 30 ml/min and 200 ml/min.

Table 5 shows a comparison between the experimental and numerical values of maximum pressure and maximum flow rate for the two pumps. The percentage difference between the experimental and numerical maximum pressure, $\delta P$ and the percentage difference between the experimental and numerical maximum flow rate, $\delta Q$ are also reported in Table 5.

As shown in Fig. 12 and in Table 5, the model can predict reasonably well the maximum pressure and maximum flow rate of the two pumps, just by considering the properties of the coils, of the magnets and the loss coefficients of the valves. However, the characteristics of the MSP3.3 cannot be represented using a linear relationship (similar to Eq. (1)) between the flow rate and the pressure. In order to capture the intermediate pressure/flow rate conditions, the flow inside the pump itself must be considered.

Even if the intermediate pressure/flow rate conditions are not well captured and a more complex analysis is required, the model can predict the maximum pressure and the maximum flow rates developed and it could be used to design and optimise a pump that would work at the desired conditions by just varying the magnetic properties.

### 5. Comparison with commercial pumps

Due to the large variety of pumps reported in the literature, it was decided to compare the MSP1.7 and MSP3.3 with commercial pumps with volume excluding connectors of less than 10 cm$^3$. Two input electrical power were considered for the MSP1.7: $W_{\text{el1}} = 1$ W and $W_{\text{el2}} = 2$ W. The following commercial products were considered:

1. Bartels-mikrotechnik MP6: piezoelectric pump [26];
2. Takasco RP-Q: peristaltic pump [27];
3. Curiejet PS22L: piezoelectric pump [28];
4. TCS M200: rotary pump [29];
5. HNP Mikrosysteme mzr-2521: gear pump [30].

Pump characteristics ($P_{\text{max}}$ and $Q_{\text{max}}$), maximum input power, volume, pumping power and pumping power per volume were
considered and a comparison between the commercial pumps and the MSPs is presented in Table 6.

The MSP1.7, even being one of the smallest available pumps, can achieve at 2 W similar maximum pressures to most of the analysed commercial pumps, only significantly inferior to the Mikrosysteme pump. The MSP3.3 instead can achieve larger flow rates than most of the commercially available pumps, significantly inferior only to the TCS pump.

Due to the relatively high pressures and flow rates developed, the MSP1.7 and MSP3.3 achieve some of the highest pumping power, inferior only to the TCS pump. Also when the pumping power per volume is considered, the MSPs are inferior only to the TCS pump. However, even if TCS pump can achieve very large flow rates (700 ml/min), the achieved maximum pressure could not be suitable for microfluidic applications where up to 100 kPa could be necessary to force the fluid through the microchannels [11].

As visible from Table 6, the MSPs compare well with commercially available μ-pumps and thanks to the model that can predict the pump characteristics, it could be possible to adjust the design of the pump to work at desired conditions.

6. Conclusions

A novel magnetic shuttle pumping technology that achieves class-leading pressures and flow rates and its model were presented in the paper. The pumping technology consisted of an oscillating neodymium magnet housed in a solenoid driver, while the model coupled the electromagnetic properties of the solenoid driver with the fluidic properties of the pump by just taking into account all the forces acting on the shuttle magnet.

Two pumps of different size (the MSP1.7 and MSP3.3) were considered to verify the ability of the model to predict the pump characteristics. The two pumps were experimentally characterised using a sinusoidal voltage signal at the electric power of 1 W: the MSP1.7 could achieve a maximum pressure of \( P_{\text{max}} = 43.53 \pm 0.24 \text{kPa} \) and maximum flow rate of \( Q_{\text{max}} = 46.69 \pm 0.32 \text{ml/min} \), while the MSP3.3 could achieve a maximum pressure of \( P_{\text{max}} = 217.4 \pm 0.13 \text{kPa} \) and maximum flow rate of \( Q_{\text{max}} = 205.99 \pm 0.18 \text{ml/min} \).

Simulations were carried out using the same electric power and predicted pump characteristics and efficiencies were calculated. Even without taking into account the complexity of the fluid inside the pump, the model could reasonably well predict the different characteristics and the optimal operation range of the two pumps. The MSP was also compared with commercial μ-pumps with volumes of less than 10 cm³ and it was shown that the proposed technology compared favourably with the other μ-pumps. Moreover, thanks to the model that can predict the pump characteristics, it could be possible to adjust the design of the pump to work at desired conditions.

Author statement

Valeria Nico: Conceptualisation, Methodology, Investigation, Formal analysis, Data curation, Writing – Original draft preparation. Eric Dalton: Conceptualisation, Methodology, Writing – Reviewing and Editing, Project administration, Funding acquisition.

Conflict of interest

The authors declare no conflict of interest.

Declaration of Competing Interest

The authors report no declarations of interest.

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