Chapter 3

Test bench

3.1 Introduction

To characterize flexible fluidic actuators, a test bench has been developed. Section 3.2 establishes the requirements of the test bench and Section 3.3 presents the design solution chosen for the test bench. This solution consists in a syringe-pump composed of a linear motor linked to a cylinder and Section 3.4 describes the methodology followed to select the linear motor and the cylinder. Afterwards, Section 3.5 describes the whole test bench and Section 3.6 finally concludes about the fulfilment of the requirements listed for the test bench.

Among all the actuators listed in the literature review (see Section 2.4), it has been decided to study PBAs ("Pneumatic Balloon Actuators", see Section 2.4.3) and miniaturized PPAMs ("Pleated Pneumatic Artificial Muscles", see Section 2.4.4 for the PAMs and Chapter 7 for the PPAMs). Miniaturized PPAMs will be developed and characterized in order to assess the miniaturization potential of the PPAMs while PBAs will be used to implement and study the PVFP principle. They have been chosen since they present one DOF, a simple design and are easily manufactured.
According to the PVFP principle, it is possible to determine the position of a flexible fluidic actuator and the force it develops thanks to the measurements of the pressure and of the volume of the fluid having been supplied to the actuator. Fig. 3.1 presents a PBA linked to a syringe-pump; the actuation fluid is air. The volume of fluid supplied to the actuator is considered to be the volume swept by the piston during its displacement; this swept volume is proportional to the piston displacement u and equals Su, where S is the syringe-pump cross-section. Therefore, the piston displacement u will be used for the PVFP principle instead of the swept volume.
When a displacement u is imposed to the piston, the inner pressure p_{in} increases and the PBA inflates and its free end A moves upwards. The vertical and horizontal displacements of this point are Δy_0 and Δx_0, respectively. Afterwards, keeping the piston position constant, if a weight w is hung from the PBA free end, the inner pressure p_{in} increases and the displacements Δy and Δx of the PBA free end decrease. According to the PVFP principle, knowing the values of p_{in} and u allows to determine the displacements of point A and the value of the weight w.

In conclusion, the test bench will be used to characterize the miniaturized PPAMs and to study the PVFP principle implemented on a PBA.

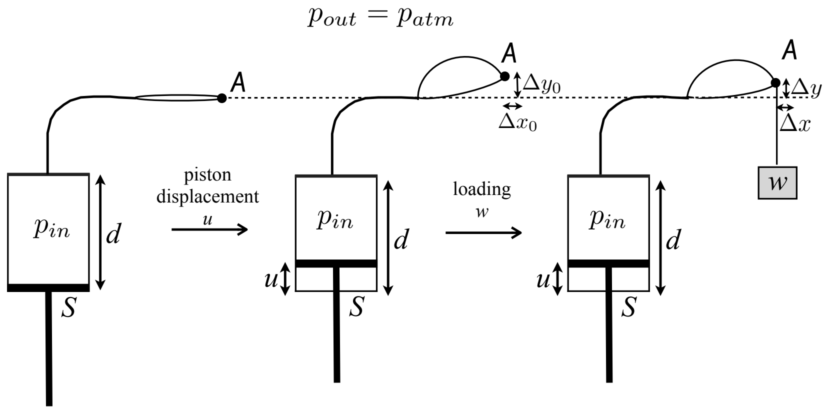

Figure 3.1: Pneumatic Balloon Actuator (PBA) linked to a syringe-pump.

3.2 Requirements

3.2.1 List of requirements

The features the test bench has to present to allow the characterization of flexible fluidic actuators and the study of the PVFP principle are the following:

- measurement of the **pressure** inside the studied flexible fluidic actuator,
- measurement of the volume of fluid supplied to the actuator: in practice, this volume will be considered to be the **volume swept** by the piston of a syringe-pump connected to the actuator,
- measurement of the **force developed** by the actuator,
- measurement of the **displacement** of the actuator,
- the test bench should be able to **maintain the pressurization states of the actuator**,
- the test bench should be able to **dynamically pressurize the actuator at a given frequency**.
- the test bench is also meant to **control the displacement and the developed force** of the studied actuators.

Also,

- the test bench should allow the characterization of a **large range of flexible fluidic actuators** and **in particular of the PBAs and miniaturized PPAMs**,
- the test bench should allow the study of flexible fluidic **actuators driven by gas or by liquid**.

3.2.2 Quantitative requirements

A) Maximum required pressurization frequency

The test bench should be able to pressurize the studied flexible fluidic actuator in 0.05 s. Indeed, a human reflex action can reach a frequency up to 10 Hz [64]. As a consequence, an actuated surgical instrument should have a bandwidth of at least 10 Hz to be able to properly follow the surgeon's movements. To test the bandwidth of the flexible fluidic actuators, the test bench should thus be able to actuate them at a rate up to 10 Hz. Hence, if the studied actuator has a bandwidth of 10 Hz, it means that it is able to go from its lower position to its upper position and back again in 0.1 s; its displacement between its extreme positions lasts 0.05 s and its pressurization should be performed in 0.05 s.

B) Ranges of the pressure, the volume of fluid supplied to the actuator, the developed force and the displacement of the studied actuator

Concerning the other requirements listed before, different application cases related to PBAs and PPAMs have been considered in order to get an idea of the pressure, volume of supplied fluid, developed force and displacement ranges of these actuators. All these cases consider that the actuation fluid is air.

- PBAs

  - The first two application cases (cases no. 0.1 and 0.2, see Table 3.1) concern a home-made PBA. It is made of two Thermoplastic Polyurethane (TPU) sheets of different thicknesses (100 µm and 200 µm). The sheets have been welded to each other with a soldering iron in order to create a square-shaped pocket presenting a cavity of 30 mm × 30 mm. The actuator is pressurized with air thanks to a syringe. When it is pressurized to the maximum, its inner volume V_{max} equals 10 ml while the syringe is completely empty. On the other hand, when the actuator is completely deflated, the volume at atmospheric pressure p_{atm} of the syringe equals $V_{atm} = 15$ ml (the volume of the actuator is assumed to equal zero). The absolute pressure p_{max} corresponding to the volume V_{max} of the pressurized actuator is then computed as follows, using the gas law for a system whose gas quantity is constant and assuming that the temperature is constant:

    $$p_{max} = \frac{p_{atm} \cdot V_{atm}}{V_{max}} \tag{3.1}$$

    The gauge pressure p_{r_max} corresponding to the absolute pressure p_{max} is computed as follows:

    $$p_{r\_max} = p_{max} - p_{atm} \tag{3.2}$$

    Besides, when pressurized and fixed as a cantilever, the home-made PBA has been able to generate at its free end a vertical force F of 0.07 N while the produced vertical displacement Δy equals 0 mm.
    All these results are summarized in the cases no. 0.1 and 0.2 in Table 3.1.
  - The three application cases no. 1.1, 1.2 and 1.3 (see Table 3.1) are based on some data found in [49]. The PBA described in [49] is a square-shaped pocket whose side length is 16 mm, while the cavity is a 10 mm × 10 mm square. The PBA is fixed as a cantilever. The maximum gauge pressure p_{r_max} reached by the actuator equals about 60 kPa (the actuation fluid is air). For this pressure, the PBA generates at its free end a vertical displacement $\Delta y = 6$ mm and a vertical force $F = 0$ N. The corresponding inner volume V_{max} of the PBA is not specified in [49] but it has been overestimated to 10 ml. Indeed, this is the volume of ten cubes of 10 mm side.
    When the PBA is pressurized at $p_{r\_max} = 20$ kPa, it can generate a vertical force $F = 0.05$ N for a vertical displacement $\Delta y = 0$ mm but it can also generate a

vertical force $F = 0$ N for a vertical displacement $\Delta y \approx 4.5$ mm.
All these results are summarized in the cases no. 1.1, 1.2 and 1.3 in Table 3.1.

- Miniature PPAMs
The PPAMs are described in details in Section 7.2.2. As the miniature PPAMs will be developed after the building of the test bench, theoretical models have been used to design the application cases related to these actuators. These theoretical models (see equations 3.3, 3.4 and 3.5) come from [30] and are presented in details in Section 7.2.2. These models assume an ideal PPAM having an inelastic membrane (see Section 7.2.2 for a definition of an ideal PPAM). They allow to estimate the equatorial diameter D and the enclosed volume V of the pressurized PPAM, as well as the pulling force F it generates:

$$D = l\ d(\epsilon, \frac{l}{R}) \tag{3.3}$$

$$V = l^3\ \nu(\epsilon, \frac{l}{R}) \tag{3.4}$$

$$F = pl^2\ f(\epsilon, \frac{l}{R}) \tag{3.5}$$

where
l = initial length of the PPAM (i.e. length of the PPAM when it is not pressurized)
l_c = contracted length of the PPAM (i.e. length of the PPAM when it is pressurized)
$\epsilon = 1 - \frac{l_c}{l}$, ϵ is the contraction of the PPAM
R = initial radius of the PPAM (i.e. radius of the PPAM when it is not pressurized)
p = gauge pressure inside the pressurized PPAM
$d(\epsilon, \frac{l}{R})$, $\nu(\epsilon, \frac{l}{R})$ and $f(\epsilon, \frac{l}{R})$ are dimensionless functions. Figures 7.5, 7.6 and 7.8 in Section 7.2.2 show the evolution of these functions with respect to the contraction ϵ and for different $\frac{l}{R}$ ratios.

For each application case, a particular muscle of initial length l and initial radius R is considered and a given contraction ϵ is assumed. The $\frac{l}{R}$ ratio and the contraction ϵ are then used to determine the values of $\nu(\epsilon, \frac{l}{R})$ and $f(\epsilon, \frac{l}{R})$. Afterwards, the enclosed volume V_{max} of the pressurized muscle is calculated using equation (3.4) while the corresponding gauge pressure p_{r_max} necessary for the muscle to reach a pulling force F of 200 N is computed thanks to equation (3.5). As explained before in Section 1.3, the miniaturized PPAMs should be able to develop a force of this order of magnitude so that once they would be integrated into a more complex medical instrument, the latter would be able to generate a force of about 13 N at its tip (see Fig. 1.5), which would allow it to perform most surgical gestures. Hence, the dimensions l and R have also been chosen so that the corresponding PPAMs would be small enough to be inserted in MIS medical instruments whose diameter must no exceed 10 mm.
The displacement Δy of the PPAM end corresponds to its contraction ϵ and is calculated as follows:

$$\Delta y = l.\epsilon \tag{3.6}$$

The quantity of gas located in the pressurized PPAM has a volume V_{max} at pressure p_{r_max} and a volume V_{atm} at atmospheric pressure p_{atm}. Hence, V_{atm} is computed as follows, using the gas law for a system whose gas quantity is constant and assuming that the temperature is constant:

$$V_{atm} = \frac{V_{max} \cdot (p_{r\_max} + p_{atm})}{p_{atm}}, \tag{3.7}$$

The application cases related to miniature PPAMs are summarized in Table 3.1. The application cases no. 2 to 5 and no. $2'$ to $5'$ assume a contraction ϵ of 10%, while cases 2" to 5" assume a contraction ϵ of 20%. For cases no. $2'$ to $5'$, the enclosed volume V_{max} is overestimated and computed as follows:

$$V_{overestimated} = \pi(\frac{D}{2})^2(l - l\epsilon) \tag{3.8}$$

with D that is calculated thanks to equation (3.3). $V_{overestimated}$ is the volume of the cylinder whose cross-section is the central cross-section of the inflated PPAM.

The last four cases of Table 3.1 (no. 6.1, 6.2, 7.1 and 7.2) concern miniature McKibben PAMs and have been found in the literature. As can be seen, the forces developed by these muscles are far lower than the forces the miniature PPAMs are expected to produce.

3.2.3 Conclusions

As explained before in Section 3.2.1, the test bench will allow to measure the pressure, the volume of supplied fluid, the developed force and the displacement of the studied flexible fluidic actuator.
As can be seen in Table 3.1, the ranges of these parameters are the following:

- pressure range (see column entitled "p_{r_max}", in Table 3.1): up to 60 kPa for the PBAs and up to 667 kPa for the PAMs. Since these ranges are very different, it will not be possible to study these two kinds of actuators with the same pressure sensor. Indeed, if the accuracy of a sensor is good enough to study the PAMs, it will not be suitable for the PBAs. Two different pressure sensors will thus be used.

- range of the volume of supplied fluid (see column entitled "V_{atm}", in Table 3.1): up to 65.75 ml.

- force range (see column entitled "F", in Table 3.1): up to 0.07 N for the PBAs and up to 200 N for the PAMs. Again, this large difference between the ranges will require a solution adapted to each of the two actuator types considered here.

- displacement range (see column entitled "Δy", in Table 3.1): up to 10 mm. The displacement Δy = 37.8 mm of the miniaturized McKibben muscle from [54] is not taken into account. Indeed, this large displacement is linked to the length (l = 180 mm) of this actuator and this length is far bigger than the one expected for the miniaturized PPAMs that will be developed.

These ranges will be used to design the test bench. However, even if these ranges are particular to PBAs and miniaturized PAMs, the test bench will allow to study other flexible fluidic actuators, as can be seen from Table 2.2.

Application case	l (mm)	R (mm)	ϵ (%)	V_{max} (ml)	p_{r_max} (kPa)	V_{atm} (ml)	F (N)	Δy (mm)
0.1 (home-made PBA)	/	/	/	10	50.65	15	0	NS
0.2				NS	NS	NS	0.07	0
1.1 (PBA, [49])	/	/	/	NS, but 10	60	14.92	0	6
1.2				NS	20	NC	0.05	0
1.3				NS	20	NC	0	≈ 4.5
2 (PPAM, $\epsilon = 10\%$)	50	5	10	18.75	106.7	38.76	200	5
3 (PPAM, $\epsilon = 10\%$)	30	2	10	4.05	404	20.41	200	3
4 (PPAM, $\epsilon = 10\%$)	20	2	10	1.2	667	9.2	200	2
5 (PPAM, $\epsilon = 10\%$)	20	4	10	2.4	526	15.02	200	2
2' (PPAM, $\epsilon = 10\%$ and $V_{overestimated}$)	50	5	10	31.81	106.7	65.75	200	5
3' (PPAM, $\epsilon = 10\%$ and $V_{overestimated}$)	30	2	10	6.87	404	34.62	200	3
4' (PPAM, $\epsilon = 10\%$ and $V_{overestimated}$)	20	2	10	2.04	667	15.65	200	2
5' (PPAM, $\epsilon = 10\%$ and $V_{overestimated}$)	20	4	10	3.62	526	22.66	200	2
2" (PPAM, $\epsilon = 20\%$)	50	5	20	25	106.7	51.68	200	10
3" (PPAM, $\epsilon = 20\%$)	30	2	20	5.4	404	27.22	200	6
4" (PPAM, $\epsilon = 20\%$)	20	2	20	1.6	667	12.27	200	4
5" (PPAM, $\epsilon = 20\%$)	20	4	20	3.2	526	20.03	200	4
6.1 (McKibben, [54])	180	0.5	21	NS	600	NC	0	37.8
6.2			0	NS	600	NC	3.8	0
7.1 (McKibben, [85])	50	3	0	NS	100	NC	10	0
7.2			0.6	NS	100	NC	0	0.3

Table 3.1: Application cases related to PBAs, miniature PPAMs and miniature McKibben PAMs. l and R are the initial length and radius of the considered miniature PAM. ϵ is the contraction presented by the muscle. V_{max} is the enclosed volume of the pressurized actuator and p_{r_max} is the corresponding gauge pressure of the actuator. The quantity of gas located in the pressurized actuator has a volume V_{max} at pressure p_{r_max} and a volume V_{atm} at atmospheric pressure p_{atm}. For the PBAs, Δy and F are the vertical displacement and force generated by the end of the actuator when the latter is fixed as a cantilever. For the PAMs, Δy is the displacement corresponding to the contraction ϵ of the muscle and F is the pulling force generated by the muscle. "NS" indicates that the data was not specified in the cited source or that the data has not been measured. "NC" indicates that V_{atm} has not been computed because V_{max} was not specified in the cited source.

3.3 Design of the test bench

A syringe-pump design is chosen for the test bench. The flexible fluidic actuator will be connected to the output of the syringe-pump and this will create a fluidic circuit composed of the chamber of the syringe-pump, the connection tubes and the flexible fluidic actuator. When the piston of the syringe-pump will be actuated, the pressure in the fluidic circuit will increase or decrease according to the actuation direction and the actuator will be actuated. In practice, a cylinder will be used as the syringe (see no. 5 in Fig. 3.2). Besides, although a syringe-pump design can be used with gas or liquid, air will be used as fluid. Indeed, this will ease the use of the test bench because air is a readily available source, it can be freely evacuated in the ambient air [70] and possible leakages will not risk damaging the test bench (e.g. the electrical connections).

The volume of fluid supplied to the actuator is considered to be the volume swept by the piston of the syringe-pump. Hence, this volume is proportional to the displacement of the cylinder piston and in practice, the piston displacement is the variable that will be used instead of the swept volume.

The solution chosen to implement the syringe-pump is presented in Fig. 3.2 and consists in the actuation of the cylinder piston (no. 4 in Fig. 3.2) by a linear motor (no. 1 in Fig. 3.2). This motor will be equipped with a position sensor to measure the displacement of the cylinder piston. To set up this solution, the linear motor only needs to be aligned with the cylinder and a part linking the cylinder piston and the slider of the linear motor (no. 3 in Fig. 3.2) needs to be manufactured.
This solution has been chosen after studying different ones. The chosen solution has been selected because most of its parts are already integrated which will ease the setting up of the test bench and because it should offer a better positioning accuracy than the other considered solutions. Section A.1 presents these other solutions and compares them to the chosen one.

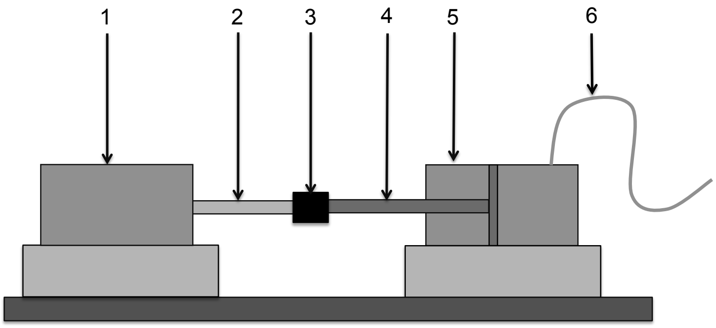

Figure 3.2: Scheme of the syringe-pump test bench: 1) linear motor + position sensor 2) slider of the linear motor 3) linking part 4) pneumatic cylinder piston 5) pneumatic cylinder 6) pneumatic tube

3.4 Pneumatic cylinder and linear motor selection

This section presents the methodology followed to select the pneumatic cylinder and the linear motor.

3.4.1 Selection of the cylinder type

As explained before, the volume of fluid supplied to the flexible fluidic actuator is considered to be the volume swept by the piston of the syringe-pump. However, considering this swept volume does make sense only if the test bench does not present any leakage. In this case, a given displacement of the cylinder piston will always lead to the same pressurization of the fluidic circuit and of the flexible fluidic actuator, when no external force is applied to the latter. Hence, a mandatory requirement for the test bench in general and for the pneumatic cylinder in particular is that there must be no leakages.
Another requirement for the test bench and the cylinder is to present as little friction as possible, because friction adds a non-linear characteristic to the system that will be difficult to model if the PVFP principle is implemented in a control loop.
Following these two requirements, the DNC cylinder with option "low friction" has been chosen among the different pneumatic cylinder types proposed by the Festo company. This cylinder presents 0.1 bar of friction. This means that at least 0.1 bar of pressure must be applied to the cylinder piston to be able to displace it. This friction is due to the sealing whose function is to insure the air-tightness of the cylinder. As long as the pressure inside the cylinder chamber remains lower than the maximum recommended pressure (12 bar), the

cylinder should not present any leakage.
The chosen cylinder has the following characteristics:

- It is a DNC cylinder with "low friction" option (= option S11), sold by the Festo company.
- It is a double-acting cylinder with a diameter of 32 mm (it is the smallest diameter available in the DNC series).
- It presents flexible cushioning rings/pads at both ends.
- Its maximum recommended speed is 0.9 m/s.

The stroke still needs to be defined and this will be done in the next section.

Remark: The Airpot Company [9] proposes pneumatic cylinders presenting less friction than those proposed by the Festo company but they have leakages.

3.4.2 Design of the "cylinder-linear motor" combination

To select a linear motor, the maximum stroke and continuous force required by the application need to be known. Besides, once the motor is chosen, a dynamic study has to be made to assess if the motor is able to perform the required movements when it is connected to its load.
To determine the maximum stroke and continuous force required for the test bench, the application cases of Table 3.1 have been considered. The goal is to use the syringe-pump test bench to inject gas in the PBAs or the miniature PPAMs of the application cases in order to actuate them.
As explained before, each application case specifies the gauge pressure p_{r_max} of the pressurized actuator as well as its enclosed volume V_{max}. To achieve this state of pressurization, the cylinder and the linear motor need a given stroke L and to maintain this state of pressurization, the linear motor must be able to develop a given continuous force $F_{max\_statics}$. These parameters have been computed for each application case with the method described below. The maximum values of these parameters have then been used to select a linear motor among the products of the Linmot company [14] and to finalize the selection of the cylinder. However, there is another requirement to take into account: as explained in Section 3.2.2, the test bench should be able to pressurize the studied actuator in 0.05 s. To assess whether the combination "cylinder-linear motor" is able to fulfil this dynamic requirement, a dynamic study has been made, as explained below.

A) Computation of the parameters L and $F_{max\_statics}$

Let us consider an inflated flexible fluidic actuator connected to a cylinder, as presented in Fig. 3.3. The enclosed volume, inner absolute pressure and inner gauge pressure of the pressurized actuator are V_{max}, p_{max} and p_{r_max}, respectively. The cylinder chamber is assumed to be empty.
To maintain the state of pressurization, the linear motor needs to develop a continuous force $F_{max\_statics}$. This force is computed as follows, by multiplying the inner gauge pressure p_{r_max} of the actuator by the cross-section S of the cylinder piston:

$$F_{max\_statics} = S \ . \ p_{r\_max}, \tag{3.9}$$

$$S = \pi(\frac{0.032 \ m}{2})^2. \tag{3.10}$$

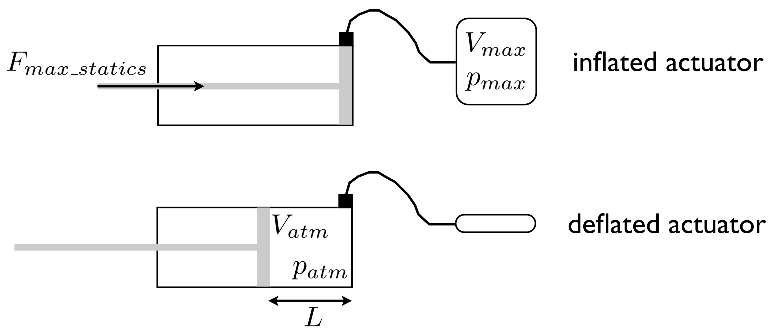

Figure 3.3: Flexible fluidic actuator connected to the output of the test bench cylinder. When inflated, the flexible fluidic actuator has an enclosed volume V_{max} and an inner absolute pressure p_{max}. To maintain the state of pressurization, the linear motor must apply a continuous force $F_{max\_statics}$ to the cylinder piston. When the actuator is completely deflated, all the gas is assumed to be located in the cylinder chamber at the atmospheric pressure p_{atm} and with a volume V_{atm}. The stroke required for the cylinder and the linear motor to pressurize the actuator is L.

In fact, equation 3.9 overestimates the force $F_{max\_statics}$ a little bit because the cylinder presents a friction of 0.1 bar. This friction contributes to maintain the piston in place but it is not taken into account in equation 3.9.
The relationship between p_{r_max} and p_{max} is the following:

$$p_{r\_max} = p_{max} - p_{atm}, \tag{3.11}$$

where p_{atm} is the atmospheric pressure.
To compute the stroke L necessary to reach such a state of pressurization, it is assumed that when the actuator is completely deflated all the gas is located in the cylinder chamber at the atmospheric pressure p_{atm} and the volume of the actuator equals zero (see Fig. 3.3). The volume of the gas at atmospheric pressure is V_{atm} and it is computed as follows, using the gas law for a system whose gas quantity is constant and assuming that the temperature is constant:

$$V_{atm} = \frac{V_{max} \cdot p_{max}}{p_{atm}} \tag{3.12}$$

V_{atm} has already been computed in Section 3.2.2 and is listed in Table 3.1.
The stroke L required to pressurize the actuator is thus:

$$L = \frac{V_{atm}}{S}, \tag{3.13}$$

Equation 3.12 assumes that the pneumatic tubes connecting the cylinder to the actuator are completely empty.

As can be seen, these simple maths allow to determine:

- the stroke L necessary to reach the pressurization state of a given flexible fluidic actuator
- the continuous force $F_{max\_statics}$ required to maintain the pressurization state.

The parameters V_{atm}, p_{atm}, V_{max} and p_{r_max} of the application cases are summarized in Table 3.2, as well as the stroke L and the continuous force $F_{max\_statics}$ required to perform each application case.

Remark: Application cases no. 0.1 and 0.2 have been removed since being able to pressurize case no. 1.1 automatically allows to pressurize both cases. Cases no. 1.2 and 1.3 have been removed for the same reason while case no. 1.1 has been kept under the number 1. Finally, cases no. 6.1, 6.2, 7.1 and 7.2 have been removed since they concern miniature McKibben muscles and because their V_{atm} parameter is not specified.

Application case	V_{atm} (ml)	p_{atm} (kPa)	V_{max} (ml)	p_{r_max} (kPa)	L (mm)	$F_{max\_statics}$ (N)
1 (PBA, [49])	15.92	101.3	10	60	19.79	48.25
2 (PPAM, $\epsilon = 10\%$)	38.76	101.3	18.75	106.7	48.19	85.81
3 (PPAM, $\epsilon = 10\%$)	20.41	101.3	4.05	404	25.38	324.92
4 (PPAM, $\epsilon = 10\%$)	9.2	101.3	1.2	667	11.44	536.43
5 (PPAM, $\epsilon = 10\%$)	15.02	101.3	2.4	526	18.68	423.04
2' (PPAM, $\epsilon = 10\%$ and $V_{overestimated}$)	65.75	101.3	31.81	106.7	81.75	85.81
3' (PPAM, $\epsilon = 10\%$ and $V_{overestimated}$)	34.62	101.3	6.87	404	43.05	324.92
4' (PPAM, $\epsilon = 10\%$ and $V_{overestimated}$)	15.65	101.3	2.04	667	19.46	536.43
5' (PPAM, $\epsilon = 10\%$ and $V_{overestimated}$)	22.66	101.3	3.62	526	28.18	423.04
2" (PPAM, $\epsilon = 20\%$)	51.68	101.3	25	106.7	64.26	85.81
3" (PPAM, $\epsilon = 20\%$)	27.22	101.3	5.4	404	33.85	324.92
4" (PPAM, $\epsilon = 20\%$)	12.27	101.3	1.6	667	15.26	536.43
5" (PPAM, $\epsilon = 20\%$)	20.03	101.3	3.2	526	24.91	423.04

Table 3.2: Parameters V_{atm}, p_{atm}, V_{max}, p_{r_max}, L and $F_{max\_statics}$ of the different application cases of Table 3.1. V_{max} is the enclosed volume of the pressurized actuator and p_{r_max} is its corresponding gauge pressure. The quantity of gas located in the pressurized actuator has a volume V_{max} at pressure p_{r_max} and a volume V_{atm} at atmospheric pressure p_{atm}. L is the necessary stroke for the cylinder and the linear motor to reach the considered state of pressurization of the actuator. $F_{max\_statics}$ is the continuous force necessary for the linear motor to maintain the considered state of pressurization of the actuator.

B) Selection of the cylinder and of the linear motor

As can be seen in Table 3.2, the longest stroke and highest continuous force necessary to perform the application cases are 81.75 mm and 536.43 N, respectively. These values have been used to select a motor among the products proposed by the Linmot company [14] and to select a DNC cylinder.
Among the proposed Linmot motors, the PO1-48X240 series is the one that can develop the highest continuous force. The value of this force is 257 N and to reach such a force, the motor needs to be installed in a flange (see Fig. A.12) and to be cooled by a fan. Although this force is approximately half of the maximum continuous force required, it will nevertheless be sufficient for the experiments to be performed.
Indeed, the application cases related to miniature PPAMs (cases no. 2 to 5, $2'$ to $5'$ and $2''$ to $5''$) consider that the actuator develops a pulling force of 200 N. However, to get interesting conclusions from the experiments, it is not necessary that the actuators reach such a force and it will be sufficient to have a test bench able to pressurize miniature PPAMs so that they develop a pulling force of 100 N. A linear motor developing a maximum continuous force of 257 N will thus be suitable.
The motor the PO1-48X240/90X240 has been finally chosen. It is able to develop its maxi-

mum peak force all over a 90 mm stroke but it has a total stroke of 240 mm. A 90 mm stroke has also been chosen for the pneumatic cylinder.
More details about the chosen motor are available in Section A.2.3.

C) Dynamic study of the "cylinder-linear motor" combination

Now that the cylinder and the linear motor have been selected, the dynamic study can be performed in order to answer the following question: is the "cylinder-linear motor" combination able to pressurize the studied actuator in 0.05 s?
To perform this study, a motor sizing software called "Linmot Designer" is used (this software is available for download on the LinMot website [14]).
To study an application case with this design software, some data must be provided such as the motor type, the servo controller type (a B1100VF controller has been chosen, see Section A.2.4 for more details about this choice), the mass the motor has to displace when it is actuated, the external force and friction applied to the motor, the motion cycle (the curve type for the displacement, the velocity and the acceleration, the displacement, the maximum recommended velocity, etc.), etc. The details of all the parameters encoded in the software are available in Section A.2.4.
If the linear motor combined with the controller and the cylinder, is not able to perform the proposed motion cycle, warnings are indicated by the Linmot Designer program. These warnings notify, for example, that the dynamic force the motor has to produce exceeds the peak force limits of the "motor - servo controller" combination.

For each application case of Table 3.2, a start position is chosen for the motor (see Section A.2.4 for more details on the start position choice) and the following motion cycle is studied:

1. a forward motion of stroke L with an external force $F_{ext} = -F_{max\_statics}$ and a maximum velocity $v_{max} = 0.9$ m/s
2. a backward motion of stroke L with an external force $F_{ext} = -F_{max\_statics}$ and a maximum velocity $v_{max} = 0.9$ m/s

The external force F_{ext} is the force applied by the pressurized gas to the cylinder piston. According to the sign convention of the software, if this force has a negative value, it means that it tends to push the slider inside the stator.
The constraint on the velocity is due to the pneumatic cylinder. Indeed, according to the Festo support, a velocity larger than 0.9 m/s could damage the flexible cushioning rings/pads placed at both ends of the chosen cylinder.

The goal of the dynamic study is to approach a duration time of 0.05 s for the forward motion and the backward motion, respectively. With this target in mind, a search is done for each application case in order to find a curve type and the values of the corresponding parameters so that the motor, combined with the controller and the cylinder, would be able to perform the motion cycle.

<u>Remark 1</u>: The available curve types for the travel through the stroke are: sine, standstill, point to point, limited jerk or minimal jerk.
<u>Remark 2</u>: In the software, the external force $F_{ext} = -F_{max\_statics}$ is applied continuously to the cylinder piston during the motion. However, in practice, when the cylinder piston is displaced and the actuator is pressurized (depressurized), the external force increases (decreases) and its intensity finally becomes equal to $F_{max\_statics}$ (0 N), at the end of the displacement. Hence, considering that $F_{ext} = -F_{max\_statics}$ during all the motion overestimates the real external force.

Table 3.3 summarizes the results of the dynamic study of the application cases of Table 3.2. For each case, the following data are listed: the parameters L and $F_{max\_statics}$ previously computed (see Table 3.2), the start position of the motor, the motion cycle (forward motion and backward motion) the motor, combined with the servo controller and cylinder, is able to perform according to the Linmot Designer program (i.e. no warnings). The curve type and its parameters are specified as well as the duration T of the motion and the external force F_{ext} applied during the motion.

As can be seen in Table 3.3, for a majority of the application cases related to miniature PPAMs, the motor is not able to pressurize the actuators when a constant external force $F_{ext} = -F_{max\_statics}$ is applied to the motor. However, for a constant external force $F_{ext} \approx -\frac{F_{max\_statics}}{2}$, the motor can perform these application cases and this is sufficient for the test bench. Indeed, as explained before, this means that the motor can pressurize the miniature PPAMs so that they can develop a pulling force of about 100 N and besides, the constant external force F_{ext} overestimates the real external force applied to the motor.
Concerning the velocity of pressurization, the pressurization times of cases no. 2, $2'$, $3'$, $2''$ and $3''$ are too long. Nevertheless, the chosen "cylinder - linear motor - controller" combination will be kept because:

- cases no. $2'$ and $3'$ consider an overestimated volume of the inflated PPAMs, this involves an overestimated stroke L and increases the time necessary to cover this distance
- the duration time of case no. $3''$ is very close to the required time
- all the application cases are overestimated since the external force F_{ext} is applied constantly during the motion whereas in reality, this force increases or decreases while the motor slider is displaced. For the motor, counterbalancing this constant external force F_{ext} consumes energy that could otherwise be used to accelerate more and decrease the duration time.

Conclusion

At the light of this dynamic study, the chosen "cylinder - linear motor - controller" combination is kept.
Table 3.4 summarizes its main properties. The datasheets of the chosen linear motor and controller are available in Section A.2.3, in Fig. A.11, A.16, A.17 and A.18.

Application case	L (mm)	$F_{max\_statics}$ (N)	start position (mm)	curve type	acc. (m/s^2)	dec. (m/s^2)	T (ms)	F_{ext} (N)
1 (PBA, [49])	19.79	48.25	25.21					
forward motion				pt 2 pt	35	35	47.6	−48.2
backward motion				pt 2 pt	35	35	47.6	−48.2
2 (PPAM, $\epsilon = 10\%$)	48.19	85.81	−3.19					
forward motion				pt 2 pt	100	100	62.5	−85.8
backward motion				pt 2 pt	100	100	62.5	−85.8
3 (PPAM, $\epsilon = 10\%$)	25.38	324.92	19.62					
forward motion				pt 2 pt	40	40	50.7	−220
backward motion				pt 2 pt	40	40	50.7	−220
4 (PPAM, $\epsilon = 10\%$)	11.44	536.43	33.56					
forward motion				pt 2 pt	20	20	47.8	−250
backward motion				pt 2 pt	20	20	47.8	−250
5 (PPAM, $\epsilon = 10\%$)	18.68	423.04	26.32					
forward motion				pt 2 pt	30	30	49.9	−240
backward motion				pt 2 pt	30	30	49.9	−240
2' (PPAM, $\epsilon = 10\%$ and $V_{overestimated}$)	81.75	85.81	−36.75					
forward motion				pt 2 pt	110	110	99	−85.8
backward motion				pt 2 pt	110	110	99	−85.8
3' (PPAM, $\epsilon = 10\%$ and $V_{overestimated}$)	43.05	324.92	1.95					
forward motion				pt 2 pt	50	50	65.8	−200
backward motion				pt 2 pt	50	50	65.8	−200
4' (PPAM, $\epsilon = 10\%$ and $V_{overestimated}$)	19.46	536.43	25.54					
forward motion				pt 2 pt	30	30	50.9	−240
backward motion				pt 2 pt	30	30	50.9	−240
5' (PPAM, $\epsilon = 10\%$ and $V_{overestimated}$)	28.18	423.04	16.82					
forward motion				pt 2 pt	50	50	49.3	−200
backward motion				pt 2 pt	50	50	49.3	−200
2" (PPAM, $\epsilon = 20\%$)	64.26	85.81	−19.26					
forward motion				pt 2 pt	110	110	79.6	−85.8
backward motion				pt 2 pt	110	110	79.6	−85.8
3" (PPAM, $\epsilon = 20\%$)	33.85	324.92	11.15					
forward motion				pt 2 pt	50	50	55.6	−200
backward motion				pt 2 pt	50	50	55.6	−200
4" (PPAM, $\epsilon = 20\%$)	15.26	536.43	29.74					
forward motion				pt 2 pt	25	25	49.4	−250
backward motion				pt 2 pt	25	25	49.4	−250
5" (PPAM, $\epsilon = 20\%$)	24.91	423.04	20.09					
forward motion				pt 2 pt	40	40	50.2	−220
backward motion				pt 2 pt	40	40	50.2	−220

Table 3.3: Dynamic study of the application cases: the table lists the motion cycles (forward motion and backward motion) that the motor, combined with the servo controller and the cylinder, is able to perform according to the Linmot Designer software. During a cycle, an external force F_{ext} is applied to the motor slider and the motor makes a forward motion and a backward motion of stroke L without exceeding the maximum velocity $v_{max} = 0.9$ m/s. The start position of the motor is specified as well as the curve type of the motion ("pt 2 pt" means "point to point") and its parameters (the acceleration ("acc."), the deceleration ("dec.")). T is the duration of the motion. L and $F_{max\_statics}$ are the parameters computed previously in Table 3.2.

Pneumatic cylinder	
brand	Festo
type	DNC-32-90-p-S11 (double-acting cylinder)
diameter	32 mm
stroke	90 mm
maximum recommended speed	0.9 m/s
friction	10 kPa
damping	flexible cushioning rings/pads placed at both ends of the cylinder
piston mass	243 g
pneumatic connexion	G1/8 female
Linear motor	
brand	Linmot
type	PO1-48X240/90X240
maximum stroke	240 mm
stroke SS	90 mm
maximum continuous force with fan cooling	257 N
force constant	39 N/A
maximum peak force (with a B1100VF controller: i.e. maximum current 8 A)	39 N/A . 8 A = 312 N
position repeatability	±0.05 mm
linearity	±0.2 %
position repeatability with external sensor	±0.01 mm
linearity with external sensor	±0.01 mm
Controller	
brand	Linmot
type	B1100-VF
maximum peak current delivered to the motor	8 A
command types	current command or slider position command

Table 3.4: Main properties of the pneumatic cylinder, the linear motor and the controller chosen for the test bench.

3.5 Description of the test bench

3.5.1 Test bench board

Most parts of the test bench are installed on a board (see Fig. 3.4). This is a M-TD-22 Breadboard proposed by Newport. Its dimensions are 600 mm × 600 mm × 28 mm. It presents threaded M6 holes distributed every 25 mm in two perpendicular directions, over the surface of the board.

Figure 3.4: Newport board on which most components of the test bench are installed. Figure from [19].

3.5.2 Platform for measurement and control

As explained in Section 3.2.1, the test bench will allow to characterize flexible fluidic actuators and to investigate the PVFP principle. To do so, it will be equipped with different sensors whose measurements will need to be acquired. Besides, the test bench is also meant to control the displacement and the developed force of the studied actuators. Hence, to acquire the sensors measurements and to offer control possibilities to the test bench, an integrated platform for measurement and control has been chosen.
National Instruments [18] and dSPACE [12] propose such platforms that have already been used by colleagues. The products proposed by these companies offer quite similar services however:

- If two similar solutions, National Instruments and dSPACE, are compared, it can be noted that the National Instruments solution will be half as expensive as the other one.
- The products proposed by National Instruments are developed for the industry. Hence, National Instruments provides follow up for its products (e.g. it proposes software updates and ensures compatibility for the components) for several tens of years. Concerning dSPACE, the products are especially developed for the automobile industry where the projects last approximately four years.
  NB: Answering questions about the lasting of a given product, a dSPACE employee said he couldn't guarantee how long the product would be proposed and would receive a follow up.

These two arguments have led to choose the National Instruments products and among the proposed platforms, the NI PXI system has been selected (see (a) in Fig. 3.5).

It uses the Labview software and is composed of a chassis, a controller and different modules that can be chosen according to the needs of the application. This makes this solution very polyvalent. The main properties of the chosen NI PXI configuration are summarized in

Main components of the chosen NI PXI configuration
Chassis with 8 slots and universal AC power supply
NI PXI-8106 controller: core 2 duo, 2.16 GHz, Windows Vista
Extra memory: 1 GB DDR2 RAM
Module with 32 analog output channels
Data acquisition module: 32 analog inputs, 48 digital inputs/outputs
Connector blocks for the analog output module and the data acquisition module

Table 3.5: Main components of the chosen NI PXI configuration

Table 3.5. A detailed list of the platform components is available in Section A.4. As can be seen in Table 3.5, "connector blocks" (see (b) in Fig. 3.5) are used. They are connected to the modules of the NI PXI platform and offer screw terminals for easy connections between the modules and the points where signals need to be sent or collected.

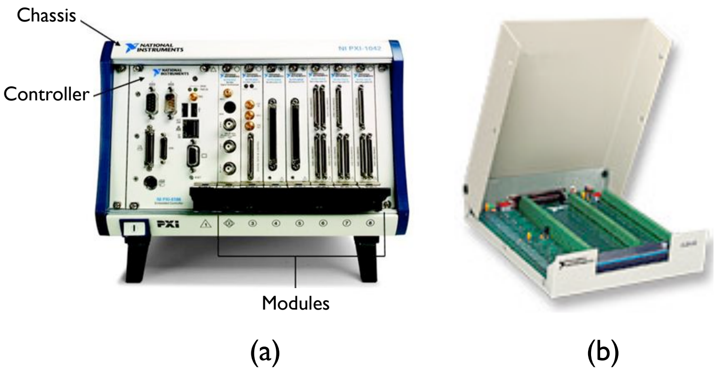

Figure 3.5: (a) NI PXI solution proposed by National Instruments (b) Connector block. Figures from [18].

3.5.3 Pneumatic cylinder and linear motor

As explained before, the test bench is composed of a linear motor combined with a pneumatic cylinder (see Fig. 3.7). This constitutes a syringe-pump that is connected to the flexible fluidic actuator to be studied. When the motor (no. 1 in Fig. 3.7) is actuated, its slider (no. 2 in Fig. 3.7) is displaced. This slider drives the piston of the cylinder (no. 4 in Fig. 3.7) and this involves a pressure variation in the pneumatic circuit (i.e. the cylinder chamber, the flexible fluidic actuator and the pneumatic tubes linking the cylinder and the actuator (no. 6 in Fig. 3.7), from which results the actuation of the actuator.

Figure 3.8 presents the practical implementation of the test bench. As can be seen in this figure,

- the linear motor (no. 1 in Fig. 3.8) is fixed on two aluminium plates A and A' which are screwed to the test bench board (no. 5 in Fig. 3.8).
- the aluminum part B ensures the linking between the slider (no. 2 in Fig. 3.8) and the piston (no. 3 in Fig. 3.8). The slider and the piston are screwed to this part.

- the cylinder (no. 4 in Fig. 3.8) is fixed on two aluminium parts C and C' which are screwed to the test bench board.

- the linear motor is equipped with a flange and a fan (no. 1 in Fig. 3.8).

- the linear motor is equipped with an external position sensor (no. 6 in Fig. 3.8). The motor has an internal position sensor that allows to position the slider with a repeatability of ±0.05 mm and a linearity of ±0.2 % (see Table 3.4). However, to have the best possible accuracy, an external sensor available in the LinMot catalog has been added to the test bench. Indeed, with such a sensor, a repeatability of ±0.01 mm is achieved as well as a linearity of ±0.01 mm (see Table 3.4).
  The external sensor has to displace above a magnetic strip (no. $6'$ in Fig. 3.8) and the sensor implementation has to fulfill some requirements presented in Fig. 3.6. The connection between the sensor and part B has thus been designed with this in mind and is ensured via the rods D and D'. Hence, when these rods are horizontal, the requirements are fulfilled.
  More information about the external position sensor is available in Section A.2.3, in Fig. A.14 and A.15.

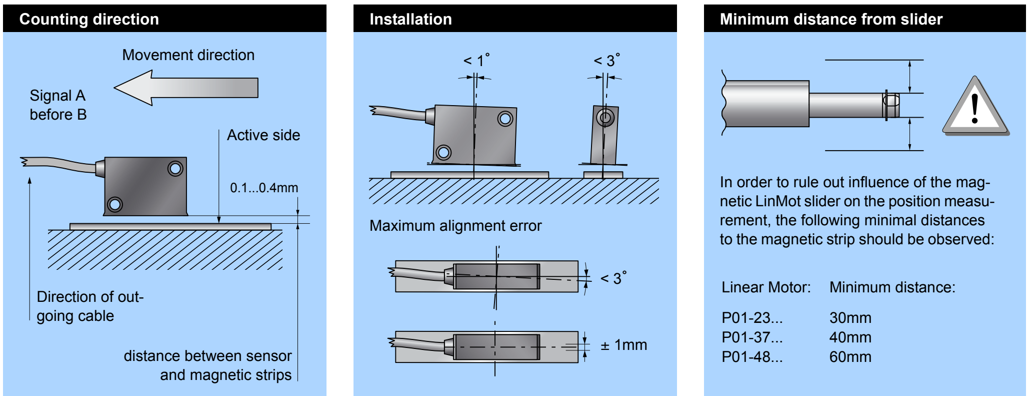

Figure 3.6: Requirements to fulfill when installing the external position sensor of the linear motor. Figure from [1].

- to ensure the horizontality of rods D and D', the rotation of the slider must be avoided. To do so, part B presents a flat surface which slides on the Teflon part E. Teflon has been chosen to limit the friction.

Remark 1: Linear guides (see Fig. A.21, in Appendix A.2.3) are available in the Linmot catalog [1]. Such guides should be used if the slider is subjected to radial loads, if the rotation of the slider or the 0.5 mm gap between the slider and the stator causes inconvenience. Hence, the Teflon part E could be replaced by a linear guide.

Remark 2: When the external position sensor is plugged to the motor controller, the latter uses the measurement of the external sensor to control the slider position, instead of using the measurement of the internal sensor of the linear motor.

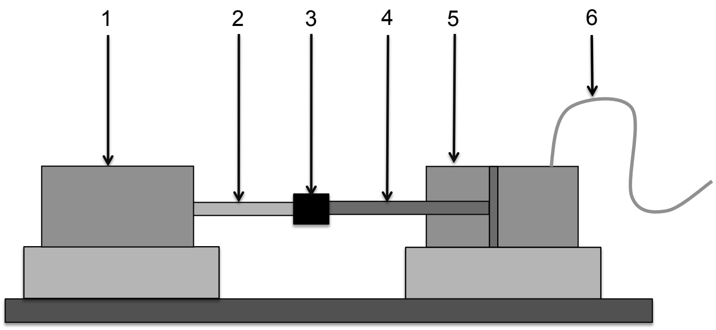

Figure 3.7: The syringe-pump test bench: 1) linear motor 2) slider of the linear motor 3) linking part 4) piston of the pneumatic cylinder 5) pneumatic cylinder 6) pneumatic tubes connecting the cylinder and the actuator (which is not represented in this figure)

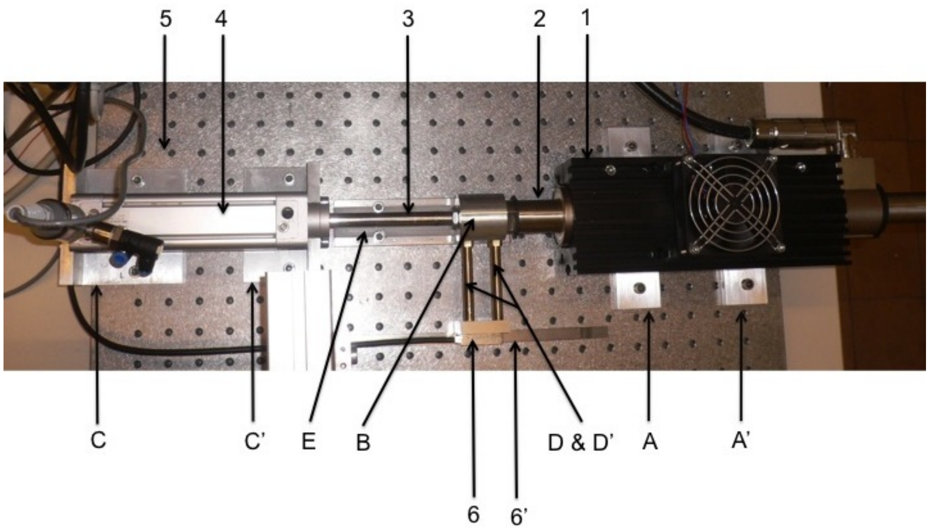

Figure 3.8: Top view of the test bench: 1) linear motor + flange + fan 2) slider of the linear motor 3) piston of the pneumatic cylinder 4) pneumatic cylinder 5) test bench board 6) external position sensor 6') magnetic strip A&A') mounting parts of the linear motor B) linking part between the slider and the piston C&C') mounting parts of the cylinder D&D') linking parts between the external position sensor and part B. E) Teflon part on which part B slides to avoid the rotation of the slider.

3.5.4 Connections between the motor controller, the motor fan and the different power supplies

Fig. A.32 in Appendix A.5 explains how the controller of the linear motor has to be connected; it is connected to two power supplies: the linear motor supply (72 VDC) and the logic supply (24 VDC). The 24 VDC logic supply was not provided with the linear motor. Hence, a 24 VDC power supply has been bought and its main characteristics are summarized in Table A.1, in Section A.5. Following the explanations of Fig. A.32, the connections between the controller, the 72 VDC motor supply, the 24 VDC logic supply, the fan and its supply have been made as shown in Fig. 3.9. The main characteristics of the fan supply (this supply was neither provided with the linear motor and has been bought separately) are summarized in Table A.2, in Section A.5.

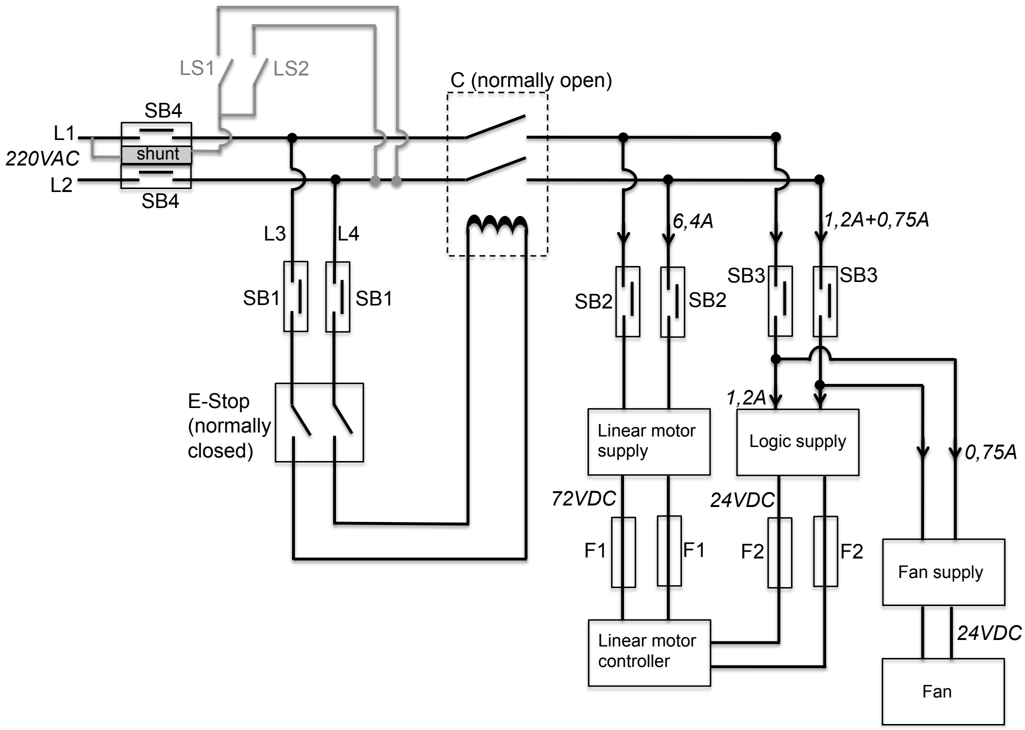

Figure 3.9: Connections between the controller, the 72 VDC motor supply, the 24 VDC logic supply, the fan and its supply. "F", "SB", "L", "C" and "LS" stand for "fuse", "switching break", "line", "contactor" and "limit switch", respectively.

As can be seen, the circuit of Fig. 3.9 presents fuses ("F" in Fig. 3.9), switching breaks ("SB" in Fig. 3.9), limit switches ("LS" in Fig. 3.9), a contactor ("C" in Fig. 3.9), a shunt and an E-stop button.
The E-Stop button allows to shut down the circuit when it is pressed (i.e. the current is switched off). As for the limit switches, they have been installed on the test bench because they can help to increase the safety of operation of the linear motor. Indeed, as the motor has a much larger stroke than the pneumatic cylinder, the limit switches can be used to shut down the motor supply when the displacement of the motor slider risks exceeding the cylinder stroke.
Fig. 3.10 presents the practical implementation of the circuit of Fig. 3.9. As can be observed, except the E-Stop button and the limit switches, all the components presented in this section are installed on two DIN rails, which are fixed to a gantry made of MiniTec parts (see Section A.3 for a short description of the MiniTec parts). Concerning the fuses, they are inserted in fuse holders that are fixed to the DIN rails.

More information about the circuit, its design and its components is available in Section A.5, as well as two summarizing tables (see Tables A.3 and A.4) presenting the main characteristics of the circuit components.

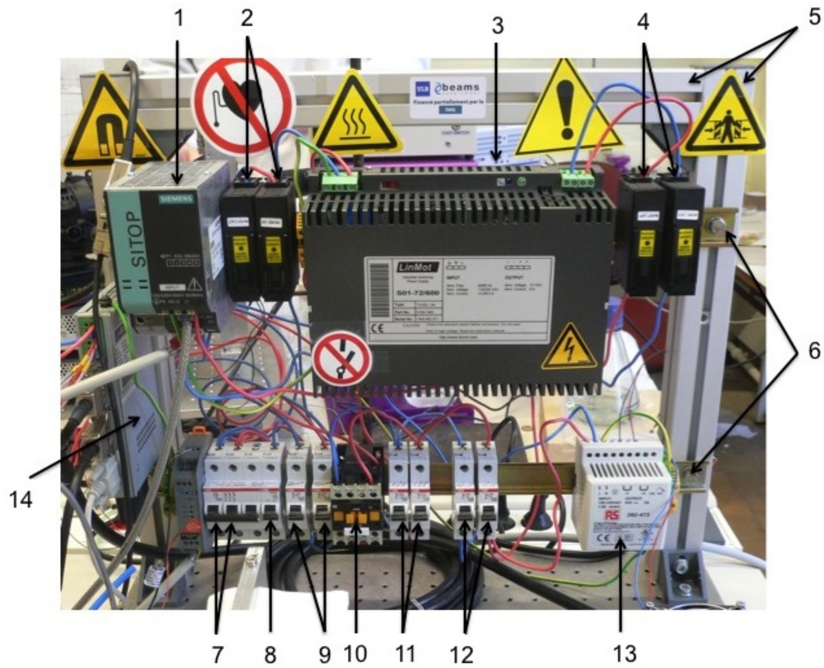

Figure 3.10: Picture of the test bench: the linear motor supply, the logic supply and the fan supply, the linear motor controller, the fuses, the switching breaks, the shunt and the contactor are installed on two DIN rails. The DIN rails are fixed to a gantry made of MiniTec parts. 1) logic supply 2) fuses F + fuse holders 3) motor supply 4) fuses F + fuse holders 5) MiniTec parts 6) two DIN rails 7) switching breaks SB4 8) shunt (bond to the SB4 switching breaks) 9) switching breaks SB 10) contactor C 11) switching breaks SB 12) switching breaks SB 13) fan supply 14) linear motor controller

Remark: In practice, the limit switches have not been used a lot. Indeed, the user-friendly program "Linmot-Talk", that enables to control the motor, allows to program the slider motion in different ways. For example, it is possible to specify a start position and an end position for the slider and to specify the velocity and acceleration of the slider when it moves between both positions. If the start and end positions are chosen so that the slider displacement does not exceed the cylinder stroke, the limit switches become superfluous.

3.5.5 Sensors

To characterize a flexible fluidic actuator, four parameters are measured for each measurement point:

1. The volume of fluid supplied to the actuator. As explained before, this volume will be considered to be the volume swept by the piston of a syringe-pump connected to the actuator. Hence, this volume is proportional to the displacement of the cylinder piston u (it is equal to the displacement of the motor slider) and in practice, the piston displacement is the variable that will be used instead of the swept volume. It is imposed by the linear motor which is equipped with a position controller and it is measured by the external position sensor of the linear motor.
   Remark: The piston displacement u is equal to the piston position with respect to the

position for which the volume of the cylinder chamber is maximum.

2. The vertical force F developed by the flexible fluidic actuator. It is measured by hanging given weights from the tip of the flexible fluidic actuator. The gravitational force of the weight corresponds to the vertical force developed upwards by the actuator.

3. The vertical displacement Δy of the actuator tip. It is measured by a camera or a laser sensor.

4. The relative pressure p inside the flexible fluidic actuator. It is measured by pressure sensors. For static measurements (i.e. all parameters are stabilized before a measurement is taken), it is assumed that the pressure is uniform in the fluidic circuit.

The measurements are made as follows:

1. The motor imposes a given displacement u_1 to the cylinder piston.

2. No external load is applied to the actuator. The pressure p and the displacement Δy are measured: their values are p_1 and Δy_1, respectively.

3. The piston position is kept constant and a given weight w is attached to the actuator tip (the corresponding force is F_2). Consequently, the pressure increases and the vertical displacement decreases. Their values are measured and are p_2 (with $p_2 > p_1$) and Δy_2 (with $\Delta y_2 < \Delta y_1$), respectively.

These three steps are then repeated for each measurement point.

A) Measurement of the piston displacement u

The piston displacement u is measured by the external position sensor of the linear motor. This sensor has a resolution of 0.001 mm and an accuracy of ± 0.01 mm/m (see Fig. A.14 and A.15, in Section A.2.3). The measurement of the sensor can be read on the control panel of the user-friendly software "Linmot-talk", which allows to control the motion of the motor. With this sensor, the motor is able to position its slider (and thus the cylinder piston) with a repeatability of ± 0.01 mm and a linearity of ± 0.01 mm (see Table 3.4, in Section 3.5.3). The volume swept by the piston is computed by multiplying the displacement of the cylinder piston by its cross-section (diameter of the cross-section = 32 mm). Hence, the accuracy of ± 0.01 mm/m on the measurement of the piston displacement involves an accuracy of ± 0.01 ml/l on the measurement of the swept volume. Besides, since the displacement of the cylinder piston can be imposed with an accuracy of ± 0.02 mm (repeatability + linearity), the swept volume can be imposed with an accuracy of ± 0.0161 ml.

B) Measurement of the actuator tip displacement Δy

The tip displacement Δy of the PBAs is measured with a Keyence camera (CV-551 controller + CV-050 camera). The controller screen presents 512 × 480 pixels (i.e. more than 240000 pixels).
The tip displacement Δy of the miniaturized PPAMs is measured with a laser sensor Keyence LC-2440. This sensor has a measurement range of ± 3 mm, a resolution of 0.2 µm and a linearity of ± 0.05%.

C) Measurement of the force F developed by the actuator

The force F developed by the flexible fluidic actuator is measured by hanging given weights at the tip of the actuator. The gravitational force of the weight corresponds to the vertical force developed upwards by the actuator.

To load the flexible fluidic actuators, KERN [13] precision weights of class M1 have been used. Some of these weights present a slot (see Fig. 3.11) and can be placed on a 10 g hook (see Fig. 3.12). The other weights (see Fig. 3.13) are already equipped with a hook. Table 3.6 presents the KERN weights and the tolerance on their values.
Dumbbell weights have also been used when heavier weights than the available KERN weights were necessary.

Figure 3.11: Picture of the KERN 10 g hook and of the KERN weights presenting a slot. These weights have been used to load the flexible fluidic actuators.

Figure 3.12: Picture of the KERN 10 g hook loaded with KERN weights presenting a slot. These weights have been used to load the flexible fluidic actuators.

Figure 3.13: Picture of the KERN weights equipped with a hook. These weights have been used to load the flexible fluidic actuators.

KERN Weight	Tolerance on the weight value
Slot weights	
1 g	± 1 mg
2 g	± 1.2 mg
5 g	± 1.6 mg
10 g	± 2 mg
20 g	± 2.5 mg
50 g	± 3 mg
100 g	± 5 mg
Hook weights	
1 g	± 1 mg
2 g	± 1.2 mg
5 g	± 1.6 mg
10 g hook	± 1.33 mg

Table 3.6: KERN weights (and the tolerance on their values) used to load the flexible fluidic actuators.

D) Measurement of the relative pressure p inside the flexible fluidic actuator

To measure the pressure inside the PBAs and the miniaturized PPAMs, different pressure sensors are used. Indeed, as explained in Section 3.2.3, these two kinds of actuators present different pressure ranges: up to 60 kPa for the PBAs and up to 667 kPa for the miniaturized PAMs, according to Table 3.1.
Hence, two relative pressure sensors presenting ranges of $0-1$ bar and of $0-6$ bar have been used for the PBAs and for the miniaturized PPAMs, respectively.
The $0-1$ bar relative pressure sensor presents the features summarized in Table 3.7. The cumulated error of this sensor has been computed on the basis of the sensor certificate of calibration. This cumulated error takes into account the accuracy, the span tolerance, the supply voltage sensitivity and equals maximum ± 0.6 % of the measured pressure value.

$0-1$ *bar* relative pressure sensor	
brand	GEMS
type	2200 R G A10 01 D 3 D B
pressure range	$0-1$ bar
zero tolerance	1 % FS[1]
pressure port	G1/4 male
response time	0.5 ms
cumulated error	max ± 0.6 % of the measured pressure value

Table 3.7: Characteristics of the $0-1$ bar relative pressure sensor

The $0-6$ bar relative pressure sensor presents the features summarized in Table 3.9. Its accuracy equals maximum ± 0.25% FS, i.e. ± 0.015 bar.
To measure the pressure inside the PBAs, a differential pressure sensor is used in addition to the $0-1$ bar relative pressure sensor. While the relative pressure sensor is used to measure the actuator pressure corresponding to a given piston position, the differential pressure sensor is used to measure the pressure variation involved when a weight is hung to the actuator tip, for a given piston position. Indeed, it has been experimentally noticed that these pressure variations are to small to be measured by the relative pressure sensor.
The differential pressure sensor has a measurement range of 1000 Pa and presents the characteristics summarized in Table 3.8. Because the fluidic circuit presents small air leakages, the pressure variation involved when attaching a weight at the PBA free end, is measured by measuring the jump that the signal of the differential pressure sensor makes when the

weight is hung. The pressure variation is thus a difference between two measurements made by the differential pressure sensor. The non-linearity and the hysteresis of the sensor both are $\leq \pm 0.10$ % FS, i.e. $\leq \pm 1$ Pa. The cumulated error on one measurement is thus $\leq \pm 0.20$ % FS, i.e. $\leq \pm 2$ Pa. Since the pressure variation will be determined by doing the difference between two measurements, the cumulated error on the pressure variation is $\leq \pm 0.40$ % FS, i.e. $\leq \pm 4$ Pa.

1000 Pa differential pressure sensor	
brand	IMPRESS sensors and systems
type	DPS 100 differential pressure transmitter 200-0100-2-3-B -Y00-1
pressure range	$0 - 1000$ Pa
non-linearity (BFSL)	$\leq \pm 0.10$ % FS
hysteresis	$\leq \pm 0.10$ % FS
pressure ports	11 mm $\times$ 6.6 mm OD for flex. tube with an $ID = 6$ mm OD=outer diameter ID=inner diameter
response time	T_{90} approx 0.02 s
cumulated error	$\leq \pm 0.40$ % FS

Table 3.8: Characteristics of the 1000 Pa differential pressure sensor

$0 - 6$ bar relative pressure sensor	
brand	Druck
type	PMP 1400
pressure range	$0 - 6$ bar
accuracy (combined non-linearity, hysteresis and repeatability) Best straight line definition	max $\pm$ 0.25% FS
zero tolerance	$\pm$ 0.5% FS
pressure port	G1/4 female
response time	/ (not specified)

Table 3.9: Characteristics of the $0 - 6$ bar relative pressure sensor

The measurements of the relative and differential pressure sensors are acquired by the NI PXI platform at a sampling rate $f_{sampling}$. Hence, anti-aliasing filters have been designed to avoid the aliasing of the acquired signals due to the sampling of the data. Indeed, according to the Shannon theorem, if a sampling rate $f_{sampling}$ is used, all the acquired signals having a frequency larger than $f_{sampling}/2$ will be aliased. To avoid this phenomenon, the sampling rate $f_{sampling}$ must verify the following condition:

$$f_{sampling} > 2 f_{max} \tag{3.14}$$

where f_{max} is the maximum frequency of the acquired signal. In practice, the following condition is applied:

$$f_{sampling} \geq 10 f_{max} \tag{3.15}$$

Since, it has been chosen to acquire the measurements with a sampling rate of 10 kHz, the maximum frequency of the acquired signal must verify the following condition:

$$1\ kHz \geq f_{max} \tag{3.16}$$

To fulfill this condition, anti-aliasing filters presenting a cutoff frequency of 1 kHz have been designed and the measurements of the pressure sensors pass through these filters before being acquired by the NI PXI platform. The electronic circuit of these filters is presented in Fig. 3.14:

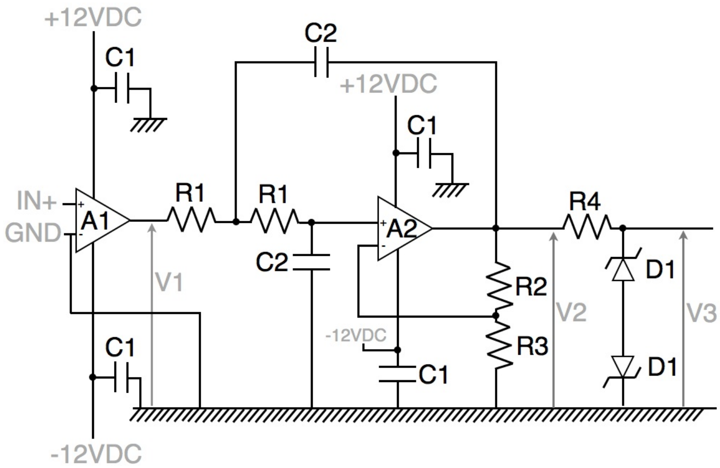

Figure 3.14: Electronic circuit of the anti-aliasing filters

This circuit is composed of:

- an instrumentation amplifier (component A1 in Fig. 3.14): it computes the difference between the two input signals. The gain of this amplifier equals 1. This amplifier has been added to the circuit in order to have an anti-aliasing filter that could be used for floating sensors as well as for referenced sensors. Indeed, this amplifier offers a differential input to the circuit.
  In practice, all the sensors of the test bench are referenced sensors whose reference is the same as the reference of the rest of the circuit. "IN+" and "GND" are the measurement signal of the sensor and the reference of the circuit, respectively.

- a second order Butterworth low-pass filter (= components located between voltages V1 and V2, A2 is an operational amplifier). Strictly speaking, this filter is the anti-aliasing filter. It has been designed so that its cutoff frequency equals 1 kHz. Its gain equals 1.586.

- two Zener diodes (components D1 in Fig. 3.14): they insure that the voltage V3, which is acquired by the NI PXI platform, stays in the range allowed by this platform, i.e. between -10 V and $+10$ V.

- decoupling capacitors (components C1 in Fig. 3.14): they help to stabilize the supply voltages. Indeed, by providing their discharge current, they avoid that an inrush current involves a voltage drop on the supply voltage lines.

More details about the following points are given in Section A.6.

- the components of these anti-aliasing circuits (see Table A.5)
- the manufacturing of the electronic cards
- the connections between the pressure sensors, the NI PXI platform, the supplies, the anti-aliasing filters and other components

3.6 Conclusions

A test bench has been developed to characterize flexible fluidic actuators and to study the PVFP principle. This test bench presents the following features:

- it allows **to characterize two specific kinds of actuators: the PBAs and the miniaturized PPAMs**. However, this test bench can be used to study **other flexible fluidic actuators**, as can be deduced from the comparison of the summarizing tables of the literature review (see Tables 2.1 and 2.2) and of the table presenting the application cases used to design the test bench (see Table 3.1).
- it is able **to maintain the pressurization states of the studied actuator**.
- it is able **to dynamically pressurize the studied actuator at a frequency up to** 10 Hz.
- it is equipped with a measurement and control platform that can be used **to control the displacement and the developed force of the studied actuator**.
- it allows **to measure the pressure inside the studied actuator**. The study of the application cases (see Table 3.1) foresees pressure ranges of $0-60$ kPa and $0-667$ kPa for the PBAs and the miniaturized PPAMs, respectively. Pressure sensors have been chosen accordingly and Table 3.10 presents the measurement ranges and accuracies of these sensors.

Sensor range and type	Accuracy
$0-1$ bar relative pressure sensor	$\pm$ 0.6 % of the measured pressure value
$0-6$ bar relative pressure sensor	$\pm$ 0.015 bar
1000 Pa differential pressure sensor	$\leq \pm 4$ Pa

Table 3.10: Measurement ranges and accuracies of the test bench pressure sensors

- it allows **to measure the cylinder piston displacement and the volume swept by the cylinder piston**. The test bench is able to measure the piston displacement u with an accuracy of $\pm$ 0.01 mm/m. Besides, it allows to impose the piston displacement u with an accuracy of $\pm$ 0.02 mm (repeatability + linearity). The test bench is able to measure the swept volume with an accuracy of $\pm$ 0.01 ml/l. Besides, it allows to impose the swept volume with an accuracy of $\pm$ 0.0161 ml.
- it allows **to measure the force developed by the studied actuator**. To do so, calibrated weights are hung at the tip of the actuator.
- it allows **to measure the displacement of the studied actuator**. To do so, a camera or a laser sensor is used.

The test bench is built as a syringe-pump whose fluidic circuit (i.e. the chamber of the cylinder, the fluidic tubes and the flexible fluidic actuator) is closed. This actuation principle can be used to pressurize pneumatic actuators with gas but also hydraulic actuators with liquid. In practice, the linear motor can be used for all the fluids while the cylinder and the pressure sensors have to be compatible with the used fluid. If they are not, it will be necessary to replace them by compatible components.

Chapter 4

Study of the PVFP principle and of the Pneumatic Balloon Actuators: Test bench particularities

4.1 Introduction

As explained before, a simple flexible fluidic actuator was necessary to investigate the PVFP principle. Therefore, an actuator having one DOF, a simple design and which is easily manufactured was looked for among the actuators listed in the review of the literature (see Section 2.4). The actuator that has finally been chosen is the "Pneumatic Balloon Actuator" (PBA, see Section 2.4.3 and [49]).
Section 4.2.1 briefly describes the original PBA developed by [49], while Section 4.2.2 presents the PBAs manufactured for the test bench. Afterwards, the particularities of the test bench linked to the study of the PBAs are presented in Section 4.3.

4.2 The Pneumatic Balloon Actuators

4.2.1 Short description of the original Pneumatic Balloon Actuator

Fig. 4.1 presents the "Pneumatic Balloon Actuator" (PBA) invented by S. Konishi et al. [49]. This device is fixed as a cantilever and comprises two square-shaped flexible films. The upper one acts as a membrane and is a silicone rubber film while the lower one plays the role of a substrate and is a polyimide film. These films are glued to one another along their surrounding edge with a silicone rubber glue and this configuration forms a cavity. When pressurized air is introduced in this cavity, the silicone rubber film inflates without supporting any bending load (like a membrane). On the other hand, the polyimide film bends due to the moment produced by the tensile forces in the membrane. This behaviour results in a large out-of-plane vertical displacement (i.e. in the y-direction) and in a horizontal displacement (i.e. in the x-direction) of the free end of the actuator. In Fig. 4.1, one can notice the presence of silicon ribs glued below the polyimide film. They aim at preventing an unwanted swelling of the substrate and at forcing the device to bend around the ribs z-axis, in order to avoid an unwanted corner folding.

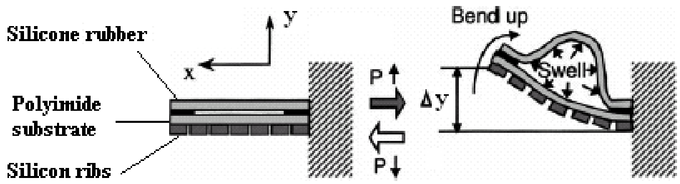

Figure 4.1: Working principle of a Pneumatic Balloon Actuator (PBA): PBA at rest and pressurized PBA on the left hand side and the right hand side, respectively. When the PBA is pressurized (i.e. the pressure P is increased), its free end moves upwards. Figure adapted from [49].

In conclusion, a PBA is an actuator presenting a cavity formed by two square-shaped flexible films of different rigidities and fastened to each other along their surrounding edge.

4.2.2 Manufacturing of Pneumatic Balloon Actuators for the test bench

As explained in Section 4.2.1, a PBA can be obtained by simply gluing two plastic squares of different rigidities along their surrounding edges, in order to form a cavity. Since this manufacturing process seemed very easy, it was decided to try developing home-made PBAs. Three methods, presented in Section B.1, have been tested: gluing the films, welding the films and moulding a PBA in latex. However, as each of these methods presented disadvantages, it has been eventually decided to look for a company to produce the PBAs. The chosen company is PRONAL [20], which is specialized in the design and manufacturing of flexible structures.
PBAs with a 40 mm × 40 mm cavity have been ordered (see Fig. 4.2).

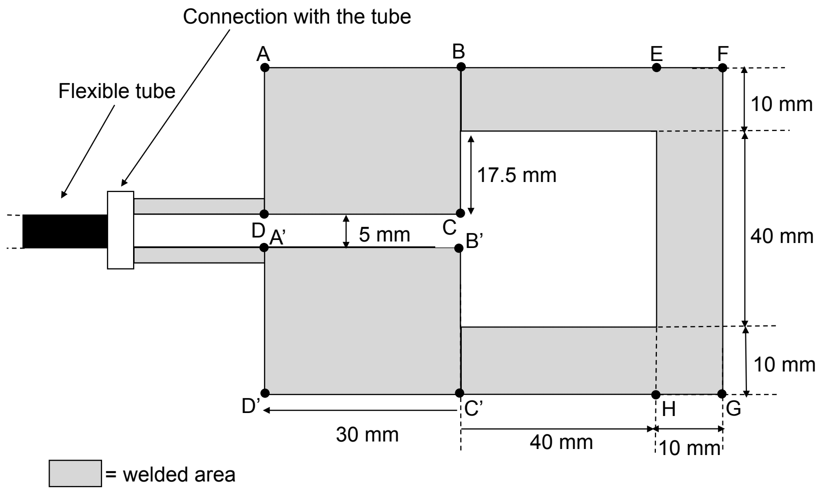

Figure 4.2: Drawing of the PBA to be manufactured by the PRONAL company. The dimensions of the cavity are 40 mm × 40 mm.

The PBAs manufactured by PRONAL are made of Polyurethane (PU), which is an elastomer. An elastomer is necessary for its elasticity, so that the films deform elastically when the PBA is pressurized and they regain their initial shape when the pressure decreases. The PBAs are obtained by fastening two layers of PU to each other by a high frequency welding

method. To have different rigidities for the two PU layers, different thicknesses have been used. Hence, the lower layer has a thickness of 1 mm and is more rigid than the upper one, which has a thickness of 0.5 mm.
The PBA should not be pressurized above $0.2 - 0.3$ bar to avoid leakages but also to avoid the plastic deformation of the polyurethane.
A flexible tube is welded to the PBA and the end of the tube is equipped with a standard pneumatic connection. The rectangles $ABCD$ and $A'B'C'D'$ (see Fig. 4.2) are aimed at being clamped to fix the PBA as a cantilever. The rectangle $EFGH$ (see Fig. 4.2) foresees room at the extremity of the PBA to hang weights for the characterization tests.
The PBAs developed by PRONAL are shown in Fig. B.7, in Appendix B.2.

Remark 1: A PBA made of two layers of the same material has a different behaviour than a PBA whose layers are of two different materials [50]. For PBAs made of only one material, when the pressure increases, their free end lifts until some pressure is reached, it then goes down when the pressure is increased further. The PBAs developed by PRONAL are entirely made of the same material and risk presenting this behaviour. However, only the lifting stage will be studied here.
Remark 2: If the PBA is placed between two plates instead of being fixed as a cantilever, it can be pressurized at pressures up to 0.5 bar, according to PRONAL.
Remark 3: PU is non-porous to air.

Fig. 4.3 presents the mounting parts used to fix the PBAs as cantilevers.

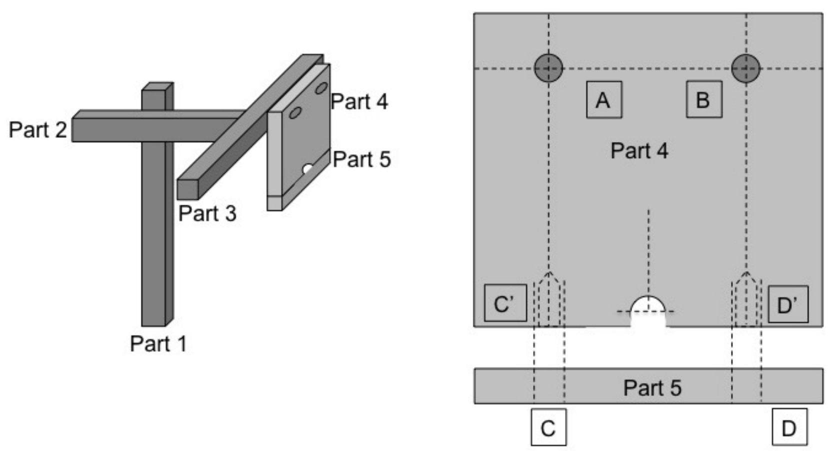

Figure 4.3: Drawings of the mounting parts used to fix the PBAs as cantilevers: the rectangles $ABCD$ and $A'B'C'D'$ of the PBA (see Fig. 4.2) are clamped between parts 4 and 5 so that the PBA is fixed as a cantilever.

Parts 1, 2 and 3 are MiniTec profiles (see Section A.3). They are assembled together with screws, nuts and squares. Part 4 is fixed to part 3 with screws, placed in the holes A and B, and with nuts. The rectangles $ABCD$ and $A'B'C'D'$ of the PBA (see Fig. 4.2) are clamped between parts 4 and 5 so that the PBA is fixed as a cantilever. Parts 4 and 5 are assembled together with screws placed in the holes C and D and in the threads C' and D'. The mounting parts allow to modify the position of the PBA in the x-, y- and z-directions.

4.3 Sensors and measurements

A PBA having a cavity of 40 mm × 40 mm is fixed as a cantilever between the parts 4 and 5 described in Section 4.2.2. The PBA is placed so that its upper layer is the thinner one. Fig. 4.4 presents a schematic view of the pressurized PBA and shows the convention chosen for the x- and y-axes. Δy and Δx are the vertical and horizontal displacements of the free end A of the PBA.

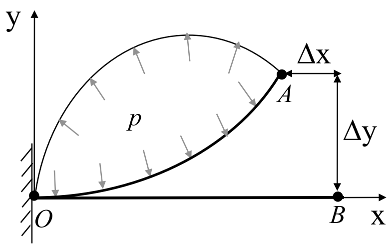

Figure 4.4: Cross-section of a pressurized PBA. The PBA is fixed as a cantilever so that its upper layer is the thinner one of its two layers. This figure shows the convention chosen for the x- and y-axes. Δy and Δx are the vertical and horizontal displacements of the free end A of the PBA.

As explained in Section 3.5.5, two different pressure sensors are used to measure the pressure inside the PBAs:

- a $0-1$ bar relative pressure sensor, whose characteristics are summarized in Table 3.7. This sensor is used to measure the actuator pressure corresponding to a given piston position.
- a $0-10$ mbar differential pressure sensor, whose characteristics are summarized in Table 3.8. This sensor is used to measure the pressure variation involved when a weight is hung to the actuator tip, for a given piston position. Indeed, it has been experimentally noticed that these pressure variations are to small to be measured by the $0-1$ bar relative pressure sensor.

The relative and differential pressure sensors are integrated into the fluidic circuit presented in Fig. 4.5. This circuit is composed of the pneumatic cylinder chamber, the flexible fluidic actuator to be studied, the pressure sensors, a solenoid valve and the fittings and tubes connecting all these components. More details about the fluidic circuit components are available in Appendix B.3.

The solenoid valve is a 2/2-way valve presenting two pressure ports; when it is open/closed, the ports are/are not connected. The solenoid valve is normally closed. This means that when it is not powered, it is closed. On the other hand, as soon as power is supplied, it commutes and opens itself. To control the closing/opening of the valve with the NI PXI platform (see Section 3.5.2), an electronic circuit has been built and it is presented in Appendix B.3.

It is assumed that the pressure is uniform in the fluidic circuit and the pressure sensors and the solenoid valve are used as follows to measure the pressure inside the PBA:

1. The solenoid valve is open and the motor moves the cylinder piston to a given position u.

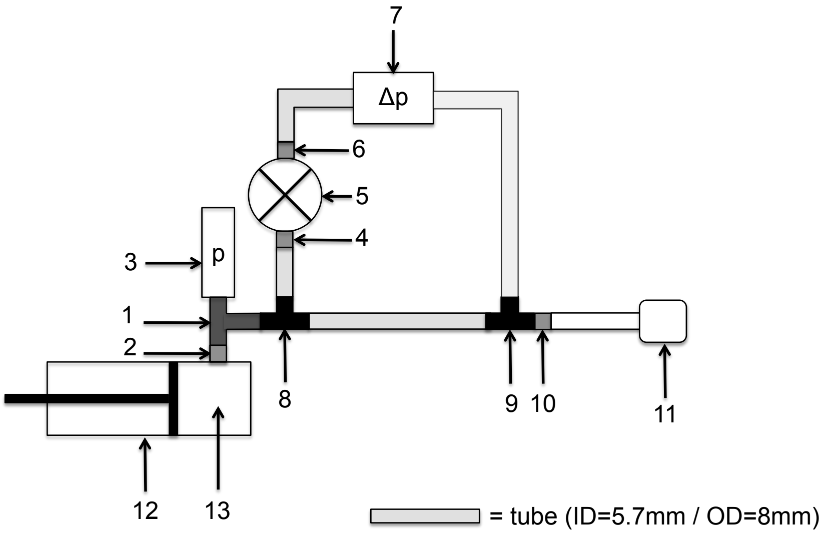

Figure 4.5: Fluidic circuit implemented to study the PBAs: 1), 2), 4), 6), 8), 9) and 10) are different kinds of fittings. 3) relative pressure sensor 5) solenoid valve 7) differential pressure sensor 11) PBA 12) pneumatic cylinder 13) pneumatic cylinder chamber. The light grey areas represent the tubes used to link the circuit components: ID=inner diameter and OD=outer diameter. All the fittings are equipped with seals.

2. No external load is applied to the PBA. The pressure p of the fluidic circuit is measured, once it is stabilized, with the relative pressure sensor. Indeed, as will be explained later, the pressure p needs some time to establish itself and to become constant.
3. The piston position u is kept constant and the solenoid valve is closed. A quantity of air at pressure p is thus trapped in the tube placed between the solenoid valve and the differential pressure sensor. A given weight is then hung to the actuator tip. Consequently, the pressure increases in the fluidic circuit (apart from the tube between the solenoid valve and the differential pressure sensor which stays at pressure p) and takes the value $p + \Delta p$. The pressure ports of the differential pressure sensor are thus at pressures p and $p + \Delta p$ and the pressure variation Δp is measured by this pressure sensor.

Hence, for a given piston position and a given weight attached to the actuator, the pressure $p + \Delta p$ of the fluidic circuit is obtained by adding the measurements of the relative and differential pressure sensors.

The X- and Y-displacements of the actuator tip Δx and Δy are measured thanks to the Keyence camera (see Section 3.5.5). A small black ball (diameter = 3 mm) is fixed at the actuator tip at 5 mm of the cavity and the position of the ball center is measured by the camera. Indeed, the camera can be programmed in order to find a given pattern in the window frame and to measure its position. Prior to making measurements with the camera, the latter needs to be calibrated in order to establish the pixels-millimeters relationship. This is done by visualizing a rectangle of graph paper in the plane where the ball placed at the actuator tip moves.

The measurements are thus made as follows:

1. The solenoid valve is open and the motor moves the cylinder piston to a given position

u.

2. No external load is applied to the PBA; the pressure p of the fluidic circuit is measured, once it is stabilized, with the relative pressure sensor and the displacements Δx_0 and Δy_0 of the actuator tip are measured with the camera.

3. The piston position u is kept constant and the solenoid valve is closed. A given weight w is then hung to the actuator tip. Consequently, the pressure increases in the fluidic circuit (apart from the tube between the solenoid valve and the differential pressure sensor which stays at pressure p) and takes the value $p + \Delta p$. The pressure variation Δp is measured by the differential pressure sensor. On the other hand, the actuator tip moves downwards and its displacements Δx and Δy are measured with the camera. The weight is then removed.

These three steps are then repeated for all the measurement points.

Remark: In practice, if the linear motor is asked to perform a given piston displacement u^*, it will perform a displacement u' close to u^* but not equal to u^*. The difference between u^* and u' can be as large as 1 mm; this differs from the positioning accuracy foreseen by the datasheets of the motor (see Section 3.5.5). Hence, in the measurements taken during the experiments, u is the piston displacement u' performed in practice and not the asked value u^*. u' is measured with the external position sensor of the motor and its measurement can be read on the control panel of the user-friendly software "Linmot-talk" (which allows to control the motion of the motor) with an accuracy of ± 0.01 mm.

The errors on the measurements are summarized in Table 4.1.

Variable	Error on the measurement of the variable
piston displacement u	± 0.01 mm
weight w	$\leq \pm$ 0.1 % of the weight (see Table 3.6, the larger error (± 0.1 % of the weight) corresponds to the tolerance on the 1 g weight)
pressure p	$\leq \pm 0.6$ % of the measured value
pressure variation Δp	$\leq \pm 4$ Pa
X- and Y- displacements Δx_0, Δy_0, Δx, and Δy	±0.13 mm

Table 4.1: Errors on the measurements

Although the test bench has been set up with a lot of care, it presents some air leakages. Between the beginning and the end of a set of experiments, the maximum drop that has noticed on the pressure p for a given piston displacement is 0.55 kPa. Hence, before performing a new set of experiments with the test bench, it is refilled with air at atmospheric pressure.