

Experimental Flow Characterization and Geometric Selection of a Passive Pneumatic Synchronizing Element

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Abstract. Passive synchronization of two pneumatic actuators is difficult because differences in external load, friction, and pneumatic resistance may cause unequal piston-rod motion. This study investigates the airflow characteristics of a passive pneumatic synchronizing element and the influence of its principal internal geometric parameters. Airflow measurements were performed at a constant supply gauge pressure of $2.4 \cdot 10^5$ Pa. Equalizing-channel diameters of $3 \cdot 10^{-4}$ m and $4 \cdot 10^{-4}$ m were tested with the annular microgap closed. The influence of the microgap height was then investigated for a constant channel diameter of $3 \cdot 10^{-4}$ m, using microgap heights from $2.5 \cdot 10^{-5}$ to $1.0 \cdot 10^{-4}$ m. Increasing the equalizing-channel diameter from $3 \cdot 10^{-4}$ m to $4 \cdot 10^{-4}$ m increased the airflow rate from $8.58 \cdot 10^{-5}$ m³/s to $1.14 \cdot 10^{-4}$ m³/s. Increasing the microgap height from $2.5 \cdot 10^{-5}$ to $1.0 \cdot 10^{-4}$ m increased the airflow rate from $8.85 \cdot 10^{-5}$ m³/s to $2.18 \cdot 10^{-4}$ m³/s. For the required airflow rate of $1.001 \cdot 10^{-4}$ m³/s, linear interpolation yielded $S_{z,int} = 3.99 \cdot 10^{-5}$ m, corresponding to approximately $4.0 \cdot 10^{-5}$ m. The airflow capacity of the passive synchronizing element can be adjusted through the combined selection of the equalizing-channel diameter, investigated at $3.0 \cdot 10^{-4}$ m and $4.0 \cdot 10^{-4}$ m, and the annular microgap height, varied from $2.5 \cdot 10^{-5}$ m to $1.0 \cdot 10^{-4}$ m. The proposed procedure provides a practical basis for adapting the element to the airflow demand and piston-rod velocity limitation of a specific dual-actuator pneumatic system. The reported values are valid for the investigated geometry and operating conditions, including a supply gauge pressure of $2.4 \cdot 10^5$ Pa, equalizing-channel diameters of $3.0 \cdot 10^{-4}$ m and $4.0 \cdot 10^{-4}$ m, and annular microgap heights from $2.5 \cdot 10^{-5}$ m to $1.0 \cdot 10^{-4}$ m.

Key words: passive pneumatic synchronization; airflow characterization; equalizing channel; annular microgap; pneumatic actuators; flow control; rehabilitation device

1. INTRODUCTION

Pneumatic actuators are widely used in industrial automation, robotic systems, assistive technologies, and rehabilitation equipment because of their relatively simple construction, low mass, favourable force-to-weight ratio, cleanliness, operational safety, and inherent mechanical compliance. Their capacity to generate smooth linear motion and transmit force through compressed air makes them particularly attractive for systems that interact directly with the human body. Pneumatic cylinders, pneumatic artificial muscles, and soft pneumatic actuators have consequently been implemented in orthopaedic rehabilitation devices, hand and wrist exoskeletons, ankle rehabilitation robots, gait-training systems, and wearable rehabilitation gloves [1], [2], [3], [4], [5].

Despite these advantages, pneumatic-drive behaviour is strongly nonlinear. The dynamic response depends on air compressibility, pressure-dependent mass flow, changing chamber volume, dead volume, tubing capacity, valve characteristics, external loading, and mechanical friction [6], [7], [8], [9]. Static friction, Coulomb friction, the Stribeck effect, seal deformation, lubrication conditions, and presliding behaviour are particularly important during motion initiation

and low-velocity operation [9], [10], [11], [12]. Service-induced changes in polyurethane seals may additionally modify friction and sealing performance, thereby contributing to differences between nominally identical actuators [12].

The simultaneous operation of two pneumatic cylinders is considerably more difficult than controlling a single actuator. Even when both cylinders have identical nominal dimensions and are supplied from the same pressure source, their piston rods may exhibit different displacement and velocity histories. Such divergence may result from unequal external loads, different seal friction, production tolerances, unequal tube lengths, local pressure losses, and differences in valve-flow characteristics [13], [14], [15], [16]. The compressibility of air allows pressure and mass-flow conditions to develop independently in each actuator branch, which may amplify relatively small differences in mechanical resistance.

This problem is particularly relevant in rehabilitation systems in which two actuators are expected to move two limbs simultaneously or symmetrically. Mechanical resistance may differ between the healthy and impaired limbs because of differences in limb mass, joint mobility, muscular stiffness, contractures, patient participation, mechanism geometry, and joint friction [1], [2], [3], [4], [5]. Pneumatic-muscle-based

mechanisms and parallel rehabilitation robots similarly demonstrate that coordinated actuator operation is essential when the intended therapeutic movement involves more than one driven segment [3], [5], [16].

In the literature, several methods have been proposed to improve the coordination of pneumatic actuators. The simplest solutions use manually adjusted throttle or flow-control valves. More advanced approaches include adaptive robust and cross-coupled control strategies [14], robust motion-control methods [17], variable-impedance control concepts [18], fuzzy control algorithms [19], friction-reduction approaches [20], and low-friction pneumatic-actuator concepts [21]. These approaches may provide high positioning accuracy and compensate for variable loading, but they also increase the number of system components, control complexity, commissioning time, and overall cost. Their application additionally requires sensors, electronic controllers, signal-processing hardware, and individual parameter tuning for each actuator branch.

Passive pneumatic synchronization offers an alternative approach in which the airflow supplied to the actuators is redistributed without continuous electronic feedback. In this type of system, pressure differences generated by unequal actuator loading produce a mechanical displacement inside the synchronizing element, which modifies the effective flow restrictions. The response is therefore generated directly by pneumatic and mechanical interactions. Such a solution may be useful in systems where simplicity, low cost, compact construction, and independence from electronic control are important.

The passive pneumatic synchronizing element was developed as a passive device for coordinating the motion of two pneumatic cylinders. Previous experimental studies demonstrated that the element can influence the velocity of two piston rods and support concurrent actuator motion under unequal loading conditions [13], [22]. The broader experimental programme, including the development of the element, the determination of its operating parameters, and tests under different actuator and loading configurations, was presented in the author's doctoral dissertation [23]. Those studies primarily evaluated piston-rod velocity, displacement histories, and the overall ability of the complete system to produce simultaneous motion. They did not, however, treat the internal airflow characteristics and the geometric selection of the flow restrictions as the principal subject of a separate investigation.

The passive pneumatic synchronizing element contains a movable spool separating two pressure chambers, annular microgaps formed between the spool and adjustable pins, equalizing channels located in the lateral spool flanges, and springs restoring the spool to its central position. A difference between the pressures in the actuator branches causes spool displacement and modifies the effective flow areas. The equalizing channels simultaneously provide a permanent flow path and reduce the possibility of complete branch closure. The performance of the element therefore depends on the interaction between the equalizing-channel diameter and the annular microgap height.

The mass-flow rate through a pneumatic restriction depends on the effective flow area, upstream and downstream absolute pressures, air temperature, discharge characteristics, and whether the flow is choked or subcritical [24], [25], [26]. Standardized procedures for determining the flow-rate characteristics of pneumatic components are provided in ISO 6358-1 [26]. In practical pneumatic systems, the experimentally determined effective area represents not only the nominal opening but also the combined influence of contraction, local pressure losses, surface condition, fittings, and internal geometry [6], [7], [8], [24], [25]. Consequently, the dimensions of the channels and microgaps cannot be selected solely from nominal actuator diameter; they must also be related to the required airflow and permissible piston-rod velocity.

Relatively small local modifications may produce measurable changes in the global response of an engineering system. The application of paint layers to a vibrating structure was shown to modify vibration velocity and shift the resonance response [27]. Changes in coating thickness and surface properties can also alter sound absorption behaviour [28], while degradation of an aircraft coating may affect aerodynamic forces and vibration levels through changes in surface condition [29]. A similar sensitivity to local geometry has been demonstrated for mechanically loaded joints. Experimental and numerical investigations of a single-lap blind-riveted joint showed that the hole-chamfer dimensions affected contact-stress distribution, rivet forming, and joint load capacity [30]. Further numerical and experimental research demonstrated that the adopted material model influenced the predicted deformation and final geometry of the rivet and rivet hole under different loading levels [31]. Although these studies concern different physical systems, they demonstrate a general engineering principle: local changes in geometry, surface condition, material representation, or resistance may propagate into measurable changes in the global system response. In the passive pneumatic synchronizing element, changes in equalizing-channel diameter on the order of $1 \cdot 10^{-4}$ m and variations in annular microgap height on the order of 10^{-5} m may substantially affect pressure loss and airflow capacity.

The geometric parameters of a passive pneumatic synchronizing element must satisfy two competing requirements. The available flow area must be sufficiently large to provide the airflow required to move the actuator under the expected external load. At the same time, excessive airflow may cause the piston-rod velocity to exceed the limit adopted for a slow rehabilitation movement and may reduce the throttling effect responsible for passive regulation. A selection criterion based only on maximum airflow would therefore be insufficient. The channel diameter and microgap height should instead be determined from both the minimum required airflow and the maximum acceptable actuator velocity.

Accordingly, the aim of this study was to experimentally determine the influence of the equalizing-channel diameter and annular microgap height on the volumetric airflow rate through a passive pneumatic synchronizing element. Measurements were conducted using a dedicated test stand at a constant supply pressure. The experimentally obtained airflow characteristics

were compared with the airflow required for a dual-actuator rehabilitation-system configuration.

The passive synchronizing element and the broader experimental programme were originally developed and documented in the author's doctoral dissertation [23], while previous journal publications focused on piston-rod velocity, displacement, and concurrent actuator motion [13], [22]. The novelty of the present study lies in the separate flow-oriented analysis of the equalizing-channel diameter and annular microgap height, the quantitative comparison of their contributions to airflow capacity, and the formulation of a geometric-selection procedure based on the required airflow and piston-rod velocity limitation.

2. EXPERIMENTAL METHODS

2.1. Passive pneumatic synchronizing element

The investigated passive pneumatic synchronizing element was developed as a passive pneumatic device for distributing compressed air between two actuator branches. The construction, operating principle, and original experimental programme of the element were previously documented in the author's doctoral dissertation [23]. Its purpose is to reduce differences in actuator motion caused by unequal external loads without using proportional valves, position sensors, or an electronic feedback controller.

The element consists of a cylindrical body containing a movable spool equipped with two lateral flanges. The spool divides the internal volume of the body into two pressure chambers connected to the corresponding pneumatic actuators. A through-channel is located along the spool axis and is supplied with compressed air through a transverse inlet channel. The incoming airflow is divided between the two sides of the element and passes through annular microgaps formed between the spool and the adjustable pins.

The body is closed at both ends by covers containing threaded pins. Springs located in the pressure chambers limit the spool displacement and restore it to its central position when the pressure difference between the two actuator branches decreases. The construction of the investigated passive pneumatic synchronizing element and its principal internal components are shown in Fig. 1.

Equalizing channels (Fig. 1) are formed in the lateral spool flanges and connect the two pressure chambers through an intermediate equalizing space.

When the loads acting on both actuators are similar, the pressures in the two chambers remain approximately balanced and the spool stays close to its central position. Under these conditions, compressed air is supplied to both actuator branches through flow paths of similar effective area.

When one actuator is subjected to a higher external load, the pressure in the corresponding branch increases. The resulting pressure difference between the two chambers displaces the spool toward the chamber with the lower pressure. This displacement changes the effective flow area of the annular

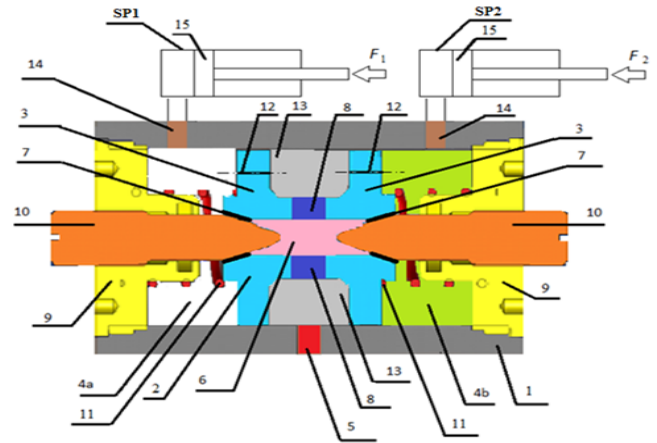


Fig. 1. Cross-sectional view of the passive pneumatic synchronizing element: 1 – body, 2 – movable spool, 3 – lateral spool flanges, 4a – left pressure chamber, 4b – right pressure chamber, 5 – inlet channel, 6 – through-channel, 7 – annular microgaps, 8 – supply channel, 9 – end cover, 10 – adjustable pin, 11 – springs, 12 – channels in the lateral spool flanges, 13 – equalizing chambers, 14 – outlet channel, SP1 and SP2 – double-acting pneumatic cylinders, 15 – cylinder piston diameter D , F_1 and F_2 – loads applied to the actuator piston rods (adapted from [23]).

microgaps and modifies the airflow distribution between the actuator branches.

The minimum pressure difference required to initiate spool displacement was not determined separately in the present experimental programme. This threshold depends on the combined influence of spring preload, friction, spool geometry, effective pressure areas, and manufacturing tolerances. Its experimental determination would therefore require a dedicated measurement of spool displacement as a function of differential pressure.

The equalizing channels in the spool flanges prevent complete closure of the flow path when the spool is displaced. Their function is therefore not only to transfer air between the pressure regions but also to prevent blockage of the more heavily loaded actuator.

The principal geometric parameters governing the airflow capacity of the passive pneumatic synchronizing element are the annular microgap height S_2 , corresponding to the annular microgaps (7), and the diameter S_3 of the equalizing channels located in the lateral spool flanges (12). The airflow supplied through the inlet channel (5) and the through-channel (6) is distributed between the annular microgap flow path (7) and the equalizing channels (12). These two geometric parameters were therefore selected for the experimental investigation presented in this study.

In accordance with the original design documentation and the doctoral dissertation [23], the symbol S_2 denotes the annular microgap height, whereas S_3 denotes the diameter of the equalizing channels in the lateral spool flanges.

The simplified model adopted in this study was used only to determine the reference operating conditions and airflow requirement. It assumes quasi-steady airflow, constant supply pressure during each test, and constant air properties over the

investigated range. Transient thermal effects and detailed local-flow phenomena were not modeled separately. The combined influence of friction, contraction, surface condition, and internal losses was accounted for implicitly through the experimentally measured airflow characteristics.

2.2. Determination of operating conditions

2.2.1. Actuator stroke and target velocity

The operating conditions adopted in the present study were derived from the design calculations and experimental programme reported in the author's doctoral dissertation [23]. The parameters relevant to the present airflow analysis are summarized below. The nominal piston stroke was selected as $L=0.4$ m. The intended duration of one extension or return movement was assumed not to exceed $t=20$ s.

The corresponding mean piston-rod velocity was calculated as:

$$v = \frac{L}{t} \quad (1)$$

which gives $v = 0.02$ m/s.

This value was treated as the upper target velocity for the considered operating condition.

2.2.2. External-load range

The external-force range was determined from the estimated mass of a human lower limb. The combined mass of the thigh, shank, and foot was calculated for body masses from 40 to 110 kg. The resulting lower-limb mass ranged from approximately 5.5 to 15.2 kg.

The corresponding gravitational force was calculated from:

$$F = m \cdot g \quad (2)$$

where:

F - gravitational force,

m - lower-limb mass,

$g = 9.81 \text{ m/s}^2$ - gravitational acceleration.

The calculated forces ranged from approximately 54 to 149 N. For the design procedure the operating load range was rounded to 50-150 N.

The maximum value of approximately 150 N was additionally confirmed in a physical test in which a physiotherapist moved a lower limb during knee flexion and extension using a force gauge.

2.2.3. Supply-pressure selection

Two pneumatic-cylinder diameters were considered in the source calculations: $D = 0.04$ m and $D = 0.032$ m.

The supply pressure was determined for the maximum design load. The calculation included an assumed actuator-efficiency coefficient of $\eta = 0.75$ and an additional allowance of 5% for internal friction. The required force was determined from:

$$F_s = \frac{F_{max}}{\eta} \quad (3)$$

followed by:

$$F_d = F_s + 0.05F_s \quad (4)$$

where:

F_d - design force.

The calculated supply gauge pressures were: $2.4 \cdot 10^5$ Pa and $3.5 \cdot 10^5$ Pa.

The experimental characterization of the passive pneumatic synchronizing element presented in this study was conducted at $p_s = 2.4 \cdot 10^5$ Pa (equivalent actuator loading pressure) corresponding to the operating condition of the 0.04 m actuator configuration.

2.2.4. Required airflow rate

The maximum required airflow rate Q_0 was determined for the operating condition corresponding to an equivalent actuator loading pressure of $p_L = 3.97 \cdot 10^5$ Pa [23]. At a supply gauge pressure of $p_s = 2.4 \cdot 10^5$ Pa, the required airflow rate was $Q_0 = 1.001 \cdot 10^{-4} \text{ m}^3/\text{s}$.

This value was adopted as the reference airflow rate for selecting the equalizing-channel diameter and annular microgap height.

2.3. Experimental Setup and Procedure

2.3.1. Test stand

The airflow characteristics of the synchronizing element were determined on the passive pneumatic synchronizing element-test stand. The stand consisted of:

1. a compressed-air source,
2. an air-preparation unit,
3. a pressure-reducing valve,
4. a pressure gauge,
5. a flowmeter,
6. the tested passive pneumatic synchronizing element.

Compressed air was supplied through the air-preparation unit. The pressure was adjusted using the reducing valve and monitored with the pressure gauge. The flowmeter was installed upstream of the synchronizing element and was used to determine the volumetric airflow rate Q_w . The arrangement of the components used during the airflow measurements is shown in Fig. 2.

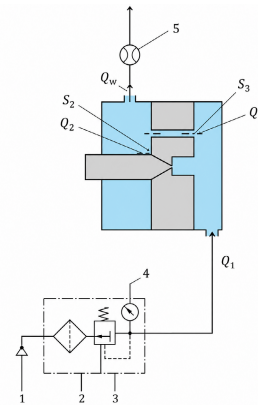


Fig. 2. Experimental setup used to determine the airflow characteristics of the passive pneumatic synchronizing element: 1 – compressed-air source, 2 – air-preparation unit, 3 – pressure-reducing valve, 4 – pressure gauge, 5 – flowmeter (adapted from [23]).

All measurements (Fig. 2) discussed in this study were carried out at a constant supply gauge pressure of $p_s = 2.4 \cdot 10^5$ Pa. The experimental study therefore focused on selecting S_2 and S_3 according to the required airflow rate of the intended pneumatic system.

For each investigated geometric configuration, six independent airflow measurements were performed. The

volumetric airflow rate was measured using an Omron D6F-50A6-000 flow sensor with a measurement range of 0-50 L/min, an accuracy of $\pm 3\%$ of full scale, and a repeatability of $\pm 0.3\%$ of full scale. The sensor output voltage was recorded using a CIE9006A digital multimeter and subsequently converted to volumetric airflow rate according to the sensor calibration characteristic. The supply pressure was adjusted and monitored using an SMC AR20-F01BE pneumatic regulator equipped with an integrated pressure gauge. The regulator has a pressure-setting range of 0.05-0.85 MPa and a maximum operating pressure of 1.0 MPa.

2.3.2. Test of the equalizing-channel diameter

In the first stage, the annular microgap was closed ($S_2 = 0$). This configuration enabled the airflow contribution of the equalizing channels to be evaluated independently. Two equalizing-channel diameters were tested: $S_3 = 3.0 \cdot 10^{-4}$ m and $S_3 = 4.0 \cdot 10^{-4}$ m. The tested values were selected on the basis of the original experimental programme reported in [23].

2.3.3. Test of the annular microgap height

In the second stage, the equalizing-channel diameter was fixed at $S_3 = 3.0 \cdot 10^{-4}$ m. The annular geometry of the adjustable microgap and the definition of its height S_2 are shown in Fig. 3.

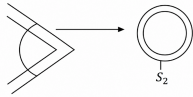


Fig. 3. Annular geometry of the adjustable microgap and definition of the microgap height S_2

For each geometry (Fig. 3), the airflow rate Q_w was recorded at the same supply pressure. The annular microgap height was varied over four levels $S_2 = 2.5 \cdot 10^{-5}$ m; $S_2 = 5.0 \cdot 10^{-5}$ m; $S_2 = 7.5 \cdot 10^{-5}$ m; $S_2 = 1.0 \cdot 10^{-4}$ m. Because the internal construction of the passive pneumatic synchronizing element was symmetrical, the airflow through one outlet branch was measured and used to characterize the corresponding microgap flow path. The experimental results were compared with the reference airflow $Q_0 = 1.001 \cdot 10^{-4}$ m³/s. The intersection between the experimentally determined $Q_w = f(S_2)$ characteristic and the reference-flow level was used to estimate the microgap height corresponding to the required airflow. The diameter $S_3 = 3.0 \cdot 10^{-4}$ m, was selected for the second experimental stage because its measured airflow with the microgap closed remained below the required reference value Q_0 , leaving a defined flow deficit to be supplemented by the annular microgap. In contrast, the $S_3 = 4.0 \cdot 10^{-4}$ m, channel already exceeded the required airflow and was therefore excluded from the microgap-selection stage.

3. RESULTS AND DISCUSSION

3.1. Influence of the equalizing-channel diameter

The experimental airflow measurements analysed in this section originate from the test programme conducted as part of the author's doctoral research [23]; the present study provides a separate flow-oriented analysis and geometric-selection framework. The airflow rates measured for the investigated equalizing-channel diameters at $p_s = 2.4 \cdot 10^5$ Pa are presented in Table 1.

Table 1. Airflow rates measured for different equalizing-channel diameters at $p_s = 2.4 \cdot 10^5$ Pa

Equalizing-channel diameter S_3 [m]	Airflow rate Q_w [m ³ /s]
$3 \cdot 10^{-4}$	$8.58 \cdot 10^{-5}$
$4 \cdot 10^{-4}$	$1.14 \cdot 10^{-4}$

Increasing the channel diameter from $3 \cdot 10^{-4}$ to $4 \cdot 10^{-4}$ m increased the measured airflow rate from: $8.58 \cdot 10^{-5}$ to $1.14 \cdot 10^{-4}$ m³/s. The relative increase was approximately:

$$\frac{1.14 \cdot 10^{-4} - 8.58 \cdot 10^{-5}}{8.58 \cdot 10^{-5}} \cdot 100\% = 32.9\% \quad (5)$$

The $3 \cdot 10^{-4}$ m channel produced an airflow approximately 14.3% lower than the required value: $Q_0 = 1.001 \cdot 10^{-4}$ m³/s. In contrast, the $4 \cdot 10^{-4}$ m channel produced an airflow approximately 13.9% higher than the required value.

The effect of the equalizing-channel diameter on the measured airflow rate is shown in Fig. 4.

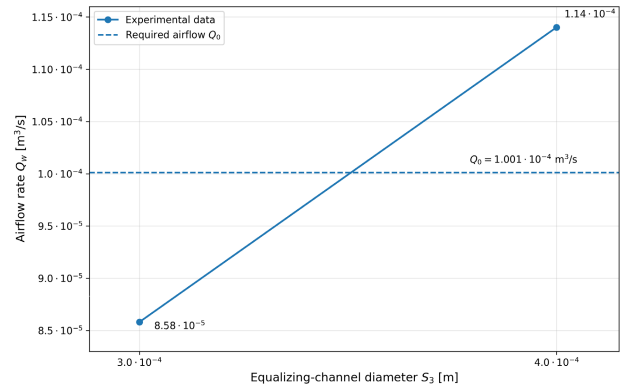


Fig. 4. Effect of equalizing-channel diameter S_3 on the airflow rate Q_w at $S_2 = 0$ and $p_s = 2.4 \cdot 10^5$ Pa. The horizontal dashed line denotes the required airflow rate $Q_0 = 1.001 \cdot 10^{-4}$ m³/s.

These results (Fig. 4.) indicate that the equalizing channel itself can provide a significant part of the total airflow capacity. A diameter of $4 \cdot 10^{-4}$ m was sufficient to exceed the reference-flow requirement even when the annular microgap was completely closed.

For this reason, the $4 \cdot 10^{-4}$ m channel was not used in the subsequent microgap-selection tests. The purpose of the next experimental stage was to determine how the missing flow

capacity of the $3 \cdot 10^{-4}$ m channel could be supplemented by an appropriately selected annular microgap.

3.2. Influence of the annular microgap height

The airflow rates obtained for the four tested microgap heights are presented in Table 2.

Table 2. Airflow rates measured for different annular microgap heights at $p_s = 2.4 \cdot 10^5$ Pa

Annular microgap height S_2 [m]	Airflow rate Q_w [m^3/s]
$2.5 \cdot 10^{-5}$	$8.85 \cdot 10^{-5}$
$5.0 \cdot 10^{-5}$	$1.08 \cdot 10^{-4}$
$7.5 \cdot 10^{-5}$	$1.62 \cdot 10^{-4}$
$1.0 \cdot 10^{-4}$	$2.18 \cdot 10^{-4}$

The results show that the airflow rate increased continuously with increasing microgap height.

When S_2 increased from $2.5 \cdot 10^{-5}$ to $5.0 \cdot 10^{-5}$ m, the airflow rate increased from $8.85 \cdot 10^{-5}$ to $1.08 \cdot 10^{-4}$ m^3/s . This represents an increase of approximately 22%. A further increase in the microgap height to $7.5 \cdot 10^{-5}$ raised the airflow rate to $1.62 \cdot 10^{-4}$ m^3/s whereas a height of $1.0 \cdot 10^{-4}$ produced $2.18 \cdot 10^{-4}$ m^3/s .

The value obtained for the largest tested microgap was more than twice the required airflow. The complete experimental characteristic and the microgap-selection criterion are shown in Fig. 5.

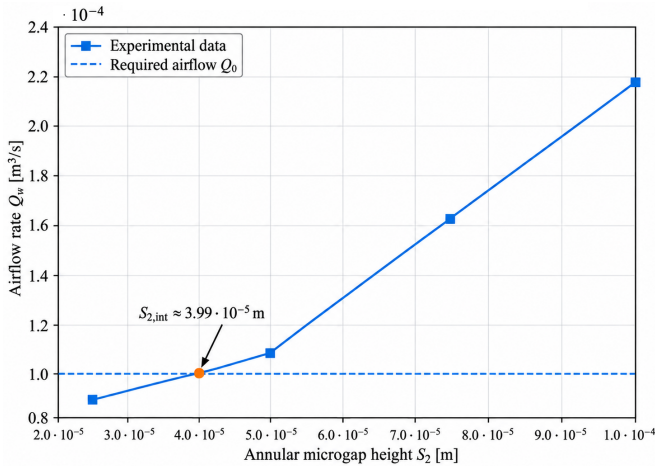


Fig. 5. Airflow rate Q_w as a function of annular microgap height S_2 at $S_3 = 3.0 \cdot 10^{-4}$ m and $p_s = 2.4 \cdot 10^5$ Pa. The horizontal dashed line denotes the required airflow rate $Q_0 = 1.001 \cdot 10^{-4}$ m^3/s , and the intersection indicates the interpolated microgap height $S_{2,int} = 3.99 \cdot 10^{-5}$ m

The experimental (Fig. 5.) trend confirms that the annular microgap provides an effective means of adjusting the flow capacity of the synchronizing element. However, the relationship was not strictly linear over the complete tested range. The increase between $2.5 \cdot 10^{-5}$ and $5.0 \cdot 10^{-5}$ m was smaller than the increases observed at higher microgap values,

which may result from the combined influence of the annular geometry, local contraction, pressure loss, and the interaction between the microgap and the equalizing channel.

3.3. Selection of the microgap height

The required airflow rate $Q_0 = 1.001 \cdot 10^{-4}$ m^3/s was located between the measured values for $S_2 = 2.5 \cdot 10^{-5}$ m and $S_2 = 5.0 \cdot 10^{-5}$ m.

Linear interpolation between the two experimental points surrounding Q_0 yielded an estimated microgap height of $S_{2,int} = 3.99 \cdot 10^{-5}$ m, corresponding to approximately $4.0 \cdot 10^{-5}$ m. This value represents the approximate microgap height at which the synchronizing element would provide the required airflow. However, the design criterion also required the piston-rod velocity not to exceed 0.02 m/s. To maintain a conservative flow limitation and prevent the actuator velocity from exceeding this value, the final microgap height was selected as $S_2 = 2.5 \cdot 10^{-5}$ m. Thus, two distinct values should be clearly distinguished:

- $S_{2,int} = 4.0 \cdot 10^{-5}$ m - the interpolated microgap height corresponding to the required airflow.
- $S_2 = 2.5 \cdot 10^{-5}$ m - the conservative final design value selected to limit the actuator velocity.

This distinction is important because the interpolated geometric value and the final engineering selection serve different purposes. The first results from matching the airflow characteristic, whereas the second includes the operational velocity constraint.

3.4. Engineering interpretation

The experiments demonstrate that both the equalizing-channel diameter and the annular microgap height contribute substantially to the total airflow capacity of the passive pneumatic synchronizing element.

The equalizing channel provides a permanent flow path and prevents complete blockage when the spool is displaced. The annular microgap acts as the adjustable regulating restriction. Therefore, the two geometric features should not be selected independently.

A larger equalizing channel:

- increases the minimum available airflow,
- reduces the risk of branch blockage,
- may decrease the relative influence of microgap regulation.

A larger annular microgap:

- increases airflow capacity,
- enables higher actuator velocity,
- may reduce the throttling action required for passive synchronization.

For the investigated operating condition the $4.0 \cdot 10^{-4}$ m equalizing channel alone exceeded the required airflow. The combination of a $3.0 \cdot 10^{-4}$ m channel and a small microgap provided a wider range for regulating the airflow. This configuration was therefore more suitable for experimental determination of the flow-limiting geometry.

The results also show that the geometric selection cannot be based only on maximizing airflow. The design should account for both:

1. the minimum airflow required to move the actuator under load.
2. the maximum acceptable actuator velocity.

This combined criterion is particularly relevant in rehabilitation-related pneumatic systems, where excessively rapid movement may be undesirable.

A practical application of the proposed selection procedure is a dual-actuator lower-limb rehabilitation device, in which two pneumatic cylinders are expected to move concurrently despite differences in external loading. In such a system, the equalizing-channel diameter and annular microgap height can be selected to provide sufficient airflow for actuator motion while maintaining the prescribed piston-rod velocity limit. This enables passive adaptation of the synchronizing element to a specific actuator size and load range without proportional valves, displacement sensors, or electronic feedback control.

The present experimental validation is limited to a supply gauge pressure of $2.4 \cdot 10^5$ Pa. Therefore, application of the proposed geometric-selection procedure at other pressure levels requires additional experimental verification.

4. CONCLUSIONS

1. Increasing the equalizing-channel diameter from $3.0 \cdot 10^{-4}$ m to $4.0 \cdot 10^{-4}$ m increased the airflow rate from $8.58 \cdot 10^{-5}$ to $1.14 \cdot 10^{-4}$ m³/s, corresponding to an increase of approximately 33%.
2. The $4.0 \cdot 10^{-4}$ m equalizing channel exceeded the required airflow even when the microgap was closed.
3. For the $3.0 \cdot 10^{-4}$ m channel, increasing the microgap height from $2.5 \cdot 10^{-5}$ to $1.0 \cdot 10^{-4}$ m increased airflow from $8.85 \cdot 10^{-5}$ to $2.18 \cdot 10^{-4}$ m³/s.
4. The interpolated microgap height corresponding to the required airflow was approximately $4.00 \cdot 10^{-5}$ m.
5. A conservative microgap height of $2.5 \cdot 10^{-5}$ m was selected to maintain the adopted piston-rod velocity limit.
6. The combined selection of equalizing-channel diameter and microgap height provides a practical method for adapting the passive element to a specific pneumatic system.

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