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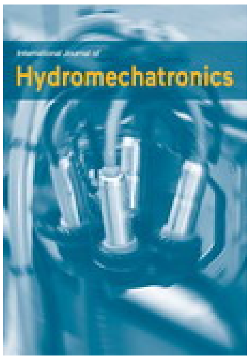

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Methods to improve motion servo control accuracy of pneumatic cylinders – review and prospect

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Abstract: Low-cost, non-polluting pneumatic technology is one of the most important engineering technologies. However, the application of pneumatic servo system is limited by the low control accuracy due to the nonlinear factors such as pneumatic actuator friction and compressibility of working medium. Many researchers have been exploring the reasons for this and have taken some targeted measures. Through extensive literature research, this paper summarises these methods into four categories: high-precision friction modelling and compensation, advanced control strategies, improved system stiffness and improved friction characteristics. In addition, the paper offers a new idea that removes the uncertain part of friction that is harmful to the control system and retains the damping part that is beneficial to the control system. For example, it can be considered to introduce a deterministic damping coefficient after removing the friction to enhance the stability of the system, and thus improve the control accuracy of the pneumatic system.

Keywords: pneumatic servo system; control accuracy; friction modelling and compensation; control strategies; system stiffness; friction characteristics; deterministic damping coefficient.

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1 Introduction

A pneumatic system is a system in which compressed air is used as the working medium to transmit energy and signals. Pneumatic systems have the benefits of low cost, simple design, cleanliness, high power to weight ratio, ease of maintenance and long-distance transmission (Kato et al., 2021; Shi et al., 2019). They also have the unique advantages such as fireproof, explosion-proof and anti-electromagnetic interference, so they have been rapidly developed and widely used. With the development of microelectronics technology and control theory, servo control technology comes into being (Fang et al., 2022; Li et al., 2022), and pneumatic servo control technology has presented a new situation of vigorous development (Wang and Liu, 2020). This makes pneumatic systems also play an important role in cutting-edge technology fields such as nuclear industry and aerospace.

Piston-type pneumatic cylinders are commonly used actuators in the pneumatic servo systems. They are often used in robotics, automation control, printing (tension control), semiconductor (spot welding machine, chip grinding) and other fields. Motion servo control and output force servo control are the two main forms of servo control for pneumatic cylinders. Among them, the motion servo control is a control means that can manage the position and speed of the cylinder moving parts in real-time to move according to the desired trajectory and the assigned parameters. In industrial applications, the motion servo control technology of pneumatic cylinders allows for flexible positioning and continuously adjustable motion speed. When the production object changes, the flexible positioning can quickly realise flexible production to meet the requirements of complex processes. The continuously adjustable motion speed can achieve the required motion speed and ideal cushioning effect. However, pneumatic servo systems have long given the impression of ‘unstable motion and poor control accuracy’. Therefore, for pneumatic technology, realising the high-accuracy motion servo control of pneumatic cylinders has always been a non-negligible area of research. At present, high-precision pneumatic cylinder positioning techniques has been investigated and reviewed in Dhavalikar et al. (2023), but the factors affecting the position control accuracy of pneumatic cylinders have not been analysed in the paper. Therefore, this paper investigates this aspect and summarises various countermeasures.

Since the normal operation of the pneumatic cylinder can not be separated from the sealing ring, it is inevitable that friction will be introduced, and the friction will be affected by the air pressure in the pneumatic cylinder. As a result, the performance of the pneumatic cylinder is less than satisfactory in occasions of low speed motion. Moreover, the compressibility of air makes the stiffness of pneumatic system very low, which increases the difficulty of the high-accuracy servo control of the pneumatic cylinder. In response to the above problems, this paper mainly investigates the measures to promote the high-accuracy motion control of pneumatic cylinders from four aspects: high-precision friction modelling (accurate friction model and precise friction test) and compensation, adopting advanced control strategies, improving pneumatic system stiffness and improving friction characteristics.

2 High-precision friction modelling and compensation

Given the unfavourable effects of friction in the pneumatic cylinders on the control performance of the systems, compensation based on friction models is a direct and effective measure for high-accuracy pneumatic servo control (Zhang et al., 2021). This requires a mathematical model that can accurately describe the friction in pneumatic cylinders. There are no uniform modelling methods for complex friction. However, with the in-depth research of scholars from various countries, the multiple behavioural characteristics of friction have gradually triggered exploration, and the friction models have been constantly perfected.

2.1 General friction model

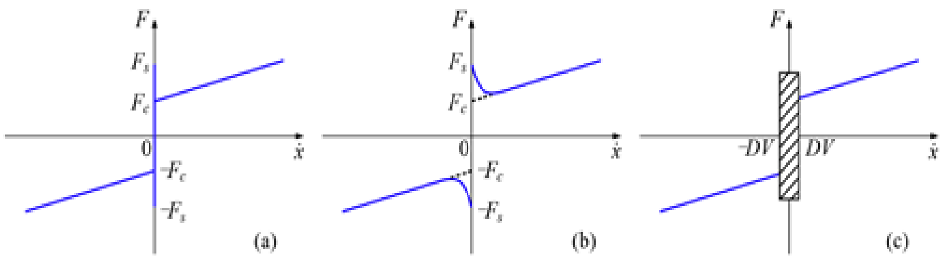
Friction models can usually be divided into static and dynamic models. The static model represents the friction as a function of speed, while the dynamic model is based on the former and can also describe the relationship between friction and displacement.

2.1.1 Static friction model

In the mid-to-late 18th century, Coulomb proposed his friction model based on the theories and experiments of Da Vinci and Amontons, that is, the friction force is proportional to the normal force on the contact surface and independent of the contact area. In 1833, Morin introduced the static friction between two contacting objects at rest and found that the static friction was higher than the Coulomb friction. In 1866, Reynolds proposed the concept of viscous friction based on the fluid viscous flow equation. At that point, the classical model of ‘static friction + coulomb friction + viscous friction’, which is most commonly used in the engineering fields, has been developed (Armstrong-Hélouvry, 1991). In the friction tests of the sliding bearings between 1900 and 1902, Stribeck found that after overcoming static friction, the friction of sliding bearings first decreased with the increase of speed, and then gradually increased (Jacobson, 2003). The combination of the Stribeck characteristics and the above friction behaviours constitutes the well-known Stribeck friction model. Whether the sudden drop between static and dynamic friction in the former model, or the negative damping characteristics in the process from static friction to dynamic friction in the latter model, it will cause the unstable ‘stick-slip’ phenomenon (creeping phenomenon) (Armstrong-Hélouvry, 1990). In 1985, the Karnopp model (Karnopp, 1985) was proposed to cope with the problem that stick-slip motion caused the need to switch back and forth between the stiction and sliding friction equations in the control processes. A zero speed range (-DV, DV) was defined in the model, where DV is the speed value that tends to zero. Outside this range, the friction is considered to be a function of speed, while within this range, the friction force depends on the other forces in the system and is less than the maximum static friction. After that, Armstrong et al. (1994) proposed a seven parameter friction model, which is a relatively complete static friction model. In the model, two equations were used to describe the friction before and during sliding, and the changing static friction and the Stribeck friction with memory effect were introduced into the equation of the sliding phase. Figure 1 shows several static models.

For the several static friction models mentioned above, we can find: Since the classical model and the Stribeck model are discontinuous at zero speed, the equation of friction needs to be switched according to the sliding speed in the control process, which requires accurate judgement of the switching speed point; Although the Karnopp model compensates for the previous drawback, the threshold value of the zero speed interval defined in the model is difficult to determine, and the friction behaviours of the model in the zero speed interval are inconsistent with the actual situation; The Armstrong model can describe most of the friction behaviour characteristics that have been observed so far, but the model has the problem of switching between the pre-sliding phase and the sliding phase, which is not reasonable from a physical point of view.

Figure 1 (a) classical friction model (b) Stribeck friction model (*Fc*–coulomb friction force, *Fs*–maximum static friction force) (c) Karnopp friction model (see online version for colours)



2.1.2 Dynamic friction model

Static friction models usually ignore the relative movement between the contact surfaces in the static friction phase, so the real friction behaviour in this phase cannot be captured. In dynamic models, the concept of pre-sliding displacement is introduced after considering the microscopic motion behaviour in the static friction stage. In 1968, Dahl (1968) first proposed the concept of pre-sliding displacement, that is, the two contact surfaces produce a tiny displacement in the static friction phase. This behaviour is similar to the deformation behaviour of a spring. In 1990, Haessig and Friedland (1990) proposed a bristle friction model by equating the contact surfaces to a large number of random contact bristles from a microscopic perspective. When the 'rigid bristles' on the sliding surface move relative to the 'elastic bristles' on the fixed surface, the contact surfaces interact like a spring, and the friction is the sum of the forces generated between each pair of contacting bristles. A reset integrator model was also mentioned in Haessig and Friedland (1990). In comparison with the bristle model, this model can not only accurately represent the microscopic stick-slip friction phenomenon, but also greatly reduce the calculation time. In 1995, Canudas de Wit et al. (1995) extended the Dahl model by combining the idea of the bristle model, and on this basis proposed the LuGre friction model. The model incorporated the Stribeck effect that the Dahl model cannot reflect, and uses the average deformation z to describe the deformation behaviour between the bristles. In 2000, Swevers et al. (2000) observed a discrepancy between the friction behaviour in the pre-sliding phase and the behaviour expressed in the LuGre model. Therefore, he improved the LuGre model by introducing a hysteresis function with non-local memory and arbitrary transition curves. Subsequently, Lampaert et al. (2002) called the friction model proposed by Swevers the Leuven model, and pointed out two improvement methods to address the shortcomings of the model. The first is to modify the nonlinear equations to eliminate the discontinuity of friction in the pre-sliding phase, and the second is to use the general Maxwell slip model to better predict the hysteresis friction. Dupont et al. (2000) proposed an elastic-plastic friction model, in which the displacement between the contact surfaces is divided into two parts: elastic displacement z and plastic displacement w. w is zero when the contact surfaces are in the stiction phase, while z is zero when the contact surfaces are in the sliding phase. Figure 2 shows several dynamic friction models.

Figure 2 (a) Dahl friction model (z–average deformation, σ_0–stiffness factor) (b) bristle friction model (F_i–force generated by the i-th pair of contacting bristles, z_i–deformation of the i-th pair of contacting bristles, b_i–position of the i-th rigid bristle, x_i–relative position of the i-th elastic bristle) (c) Lugre friction model (σ_1–microscopic damping coefficient) (d) elastic-plastic friction model (w–plastic displacement, z–elastic displacement) (see online version for colours)

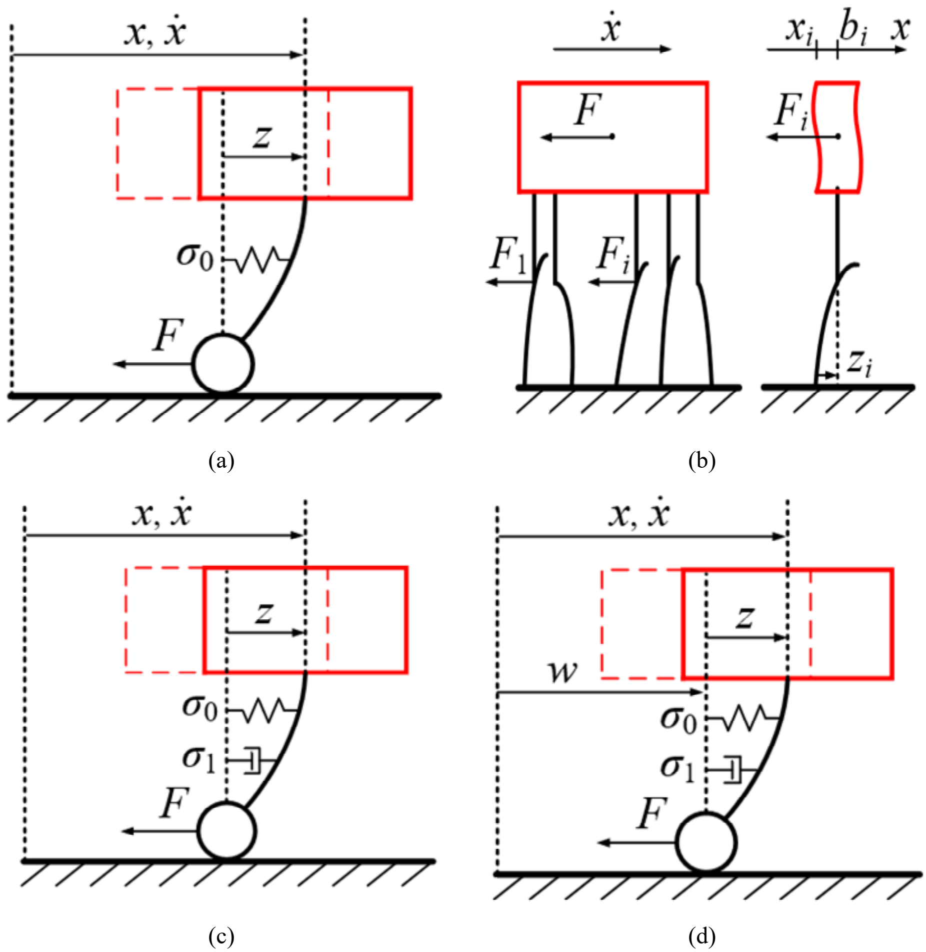


Table 1 Details of friction models

Reference	*Formula*	*Static or dynamic model*	*Friction characteristics*	*Contributions*
Armstrong-Hélouvry (1991)	$F=\begin{cases} F_e & F_e<F_s \\ F_s\,\mathrm{sgn}(F_e) & F_e\ge F_s \ \mathit{and}\ v=0 \\ F_C\,\mathrm{sgn}(v)+k_v v & F_e\ge F_s\ \mathit{and}\ v\ne 0 \end{cases}$	Static	Static friction, coulomb friction, viscous friction	Extensive engineering applications
Jacobson, B. (2003)	$F=\begin{cases} F_e & F_e<F_s \\ F_s\,\mathrm{sgn}(F_e) & F_e\ge F_s\ \mathit{and}\ v=0 \\ \left(F_C+(F_s-F_C)e^{-(v/v_S)^\delta}\right)\mathrm{sgn}(v)+k_v v & F_e\ge F_s\ \mathit{and}\ v\ne 0 \end{cases}$	Static	Static friction, coulomb friction, viscous friction, Stribeck effect	Proposed the transition from static to dynamic friction
Dahl (1968)	$\frac{dF}{dx}=\sigma_0\left\|1-\frac{F}{F_C}\mathrm{sgn}(v)\right\|^\alpha \mathrm{sgn}\left(1-\frac{F}{F_C}\mathrm{sgn}(v)\right),$ $F=\sigma_0 z,$ $\frac{dz}{dt}=v-\frac{\sigma_0\|v\|}{F_C}z$	Dynamic	Pre-sliding displacement, coulomb friction, viscous friction	Proposed the concept of pre-sliding displacement
Karnopp (1985)	$F=\begin{cases} \max(F_e,\ -F_s) & \|v\|<DV\ \mathit{and}\ F_e<0 \\ \min(F_e,\ F_s) & \|v\|<DV\ \mathit{and}\ F_e\ge 0 \\ F_C\,\mathrm{sgn}(v)+k_v v & \|v\|\ge DV \end{cases}$	Static	Static friction, coulomb friction, viscous friction	Solved the problem of friction equation switching at different speeds
Armstrong-Hélouvry et al. (1994)	$F(x)=-k_t x,$ $F(v,t)=-\left(F_C+F_S(\gamma,t_2)\frac{1}{1+(v(t-\tau_L)/v_S)^2}\right)\mathrm{sgn}(v)-k_v v,$ $F_S(\gamma,t_2)=F_{S,a}+\left(F_{S,\infty}-F_{S,a}\right)\frac{t_2}{t_2+\gamma}$	Static	Pre-sliding displacement, friction hysteresis, varying static friction, coulomb friction, viscous friction, Stribeck effect	Static friction model including most friction behaviours
Haessig and Friedland (1990)	$F=\left(1+\alpha(z)\right)\sigma_0(v)z+\sigma_1\frac{dz}{dt},$ $\frac{dz}{dt}=\begin{cases} 0 & v>0,\ z\ge z_0\ \mathit{or}\ v<0,\ z\le z_0 \\ v & \mathit{otherwise} \end{cases},$ $\alpha(z)=\begin{cases} \alpha & \|z\|<z_0 \\ 0 & \mathit{otherwise} \end{cases}$	Dynamic	Pre-sliding displacement, viscous friction	Accurately described the microscopic stick-slip friction phenomenon
Canudas de Wit et al. (1995)	$F=\sigma_0 z+\sigma_1\frac{dz}{dt}+\sigma_2 v,$ $\frac{dz}{dt}=v-\frac{\sigma_0\|v\|}{g(v)}z,$ $g(v)=F_C+\left(F_s-F_C\right)e^{-(v/v_s)^2}$	Dynamic	Pre-sliding displacement, friction hysteresis, static friction, coulomb friction, viscous friction, STRIBECK effect	Captured most friction phenomena

Table 1 Details of friction models (continued)

Reference	*Formula*	*Static or dynamic model*	*Friction characteristics*	*Contributions*
Swevers et al. (2000)	$F = F_h(z) + \sigma_1 \frac{dz}{dt} + \sigma_2 v,$ $F_h(z) = F_b + F_d(z),$ $\frac{dz}{dt} = v\left(1 - \left\lvert\frac{F_d(z)}{g(v) - F_b}\right\rvert^n \operatorname{sgn}\left(\frac{F_d(z)}{g(v) - F_b}\right)\right),$ $g(v) = F_C + (F_s - F_C)e^{-(v/v_s)^2}$	Dynamic	Pre-sliding displacement, friction hysteresis, static friction, coulomb friction, viscous friction, Stribeck effect	Introduced a hysteresis function with nonlocal memory and arbitrary transition curves
Dupont et al. (2000)	$F = \sigma_0 z + \sigma_1 \frac{dz}{dt} + \sigma_2 v,$ $\frac{dz}{dt} = v - \alpha(z, v)\frac{\sigma_0 \lvert v\rvert}{g(v)} z,$ $g(v) = F_C + (F_s - F_C)e^{-(v/v_s)^2},$ $if \;\; \operatorname{sgn}(v) = \operatorname{sgn}(z),$ $\alpha(z, v) = \begin{cases} 0 & \lvert z\rvert < z_{ba} \\ \frac{1}{2}\sin\left(\frac{z - (z_{\max} + z_{ba})/2}{z_{\max} - z_{ba}}\pi\right) + \frac{1}{2} & z_{ba} < \lvert z\rvert < z_{\max} \\ 1 & \lvert z\rvert > z_{\max} \end{cases}$ $if \;\; \operatorname{sgn}(v) \neq \operatorname{sgn}(z),$ $\alpha(z, v) = 0$	Dynamic	Pre-sliding displacement, static friction, coulomb friction, viscous friction, Stribeck effect	Avoided switch functions or discontinuities

A more detailed summary of the static and dynamic friction models described above is shown in Table 1.

In addition to the multiple dynamic friction models presented above, there are also many other different models, such as Bliman-Sorine friction model (Bliman and Sorine, 1991), Valanis friction model (Lenz and Gaul, 1995), single-state elastic-plastic friction model (Dupont et al., 2002), generalised Maxwell-Slip model (Lampaert et al., 2003), generalised friction model (Al-Bender et al., 2004a; 2004b), and single-state and multi-state integral friction models (Ferretti et al., 2004). In general, dynamic friction models have good continuity and can describe certain dynamic friction characteristics. However, the more comprehensive the model describing the friction behaviour, the more complex its structure tends to be, and the more difficult its model parameters are identified.

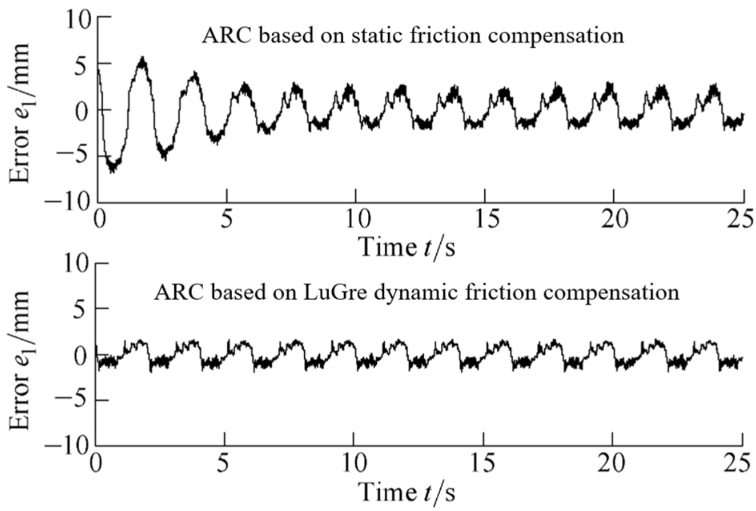
2.2 Friction model for pneumatic cylinders

Currently, many valuable friction models have been proposed and gradually improved (as described in Section 2.1). However, friction force of pneumatic cylinders is difficult to be directly represented by the existing models due to the influence of many factors such as moving directions, working pressures, standing time, ambient temperatures and humidities, shapes of seals, lubrication conditions, and material geometry errors. Researchers usually study the friction characteristics of pneumatic cylinders by simulating the actual working conditions of pneumatic systems, and on this basis, propose corresponding empirical models or further improve the existing friction models.

Belforte et al. (1989) investigated the effects of motion speed of the pistons and working pressure of the chambers on the friction characteristics of cylinders through extensive tests, and gave an empirical friction model of the pneumatic cylinders. Schroeder and Singh (1993) experimentally evaluated the accuracy of several empirical formulas for friction. Nouri et al. (2000) successfully predicted the friction characteristics of pneumatic servo positioning systems in the pre-sliding and sliding phases based on the Leuven friction model and determined the relevant model parameters. In the friction tests of the pneumatic cylinders, Cao (2009) found that the maximum static friction not only increased with the increase of stagnation time, but also had a certain relationship with the rate of pressure rise in the chambers. He finally concluded a complete nonlinear model for friction of the pneumatic cylinders.

In addition, some progress has been made in the application of the LuGre friction model to pneumatic servo systems. Chen et al. (2010) studied the effects of pressures and differential pressures on the static and dynamic parameters of the LuGre model, and obtained a complete friction model for the rodless pneumatic cylinder measured under the given working pressure conditions. Later, Meng et al. (2014a) designed an adaptive robust controller based on LuGre dynamic friction compensation to achieve trajectory tracking control of a rodless pneumatic cylinder servo system. Figure 3 showed that the controller with LuGre model-based compensation can achieve better control performance than the controller based on static friction compensation used for comparison. Tran et al. (2015) focused on observing the friction behaviour in the pre-sliding state, and established a new improved LuGre friction model in their study on the mathematical model of pneumatic cylinder friction. The new friction model can simulate well all the observed friction behaviours in both pre-sliding and true sliding states of the pneumatic cylinders.

Figure 3 Experimental results of arc based on LuGre dynamic friction compensation



Source: Meng et al. (2014a)

2.3 Friction test for pneumatic cylinders

The parameters of friction models vary greatly because the friction in the pneumatic cylinders is affected by the internal factors of cylinders themselves and the external factors of working conditions. Researchers usually use test means to directly obtain the friction force of the cylinders to be tested under actual working conditions, and identify the parameters of the models or obtain relevant empirical models for accurate friction compensation. The methods of friction tests of pneumatic cylinders can be mainly classified into the direct method and the indirect method.

2.3.1 Direct measurement method

The direct method is to directly obtain the dynamic and static friction of the cylinders according to the force information collected by the force sensors and the force balance equation. The direct measurement method requires external drive elements to provide power for the linear motion of the pneumatic cylinders. Belforte et al. (2013) placed the cylinder barrel of the pneumatic cylinder under test on a linear air-floating table connected to a fixed table through a force sensor, and used an electric cylinder to drive the piston movement. In the test, the friction force was transmitted to the force sensor through the cylinder barrel. Subsequently, they added some other components to this device, and developed a test bench that could individually measure the friction force at the rod seal and the piston seal (Belforte and Mazza, 2016). More researchers tend to fix the piston rods of pneumatic cylinders. Chang et al. (2015) used a ball screw sliding table equipped with a servo motor to guide a pneumatic cylinder at a uniform speed. The friction force was reflected in a force sensor through the piston rod. To study the sealing performance and friction characteristics of the pneumatic cylinder, Dilixati et al. (2016) built a test setup, where an electric cylinder was used to drive the reciprocating motion of a pneumatic cylinder, and the piston rod and force sensor were fixed. Wakasawa et al.

(2019) combined a stepper motor and a rack and pinion drive mechanism to provide precise speed control for the cylinder barrel of a pneumatic cylinder, and fixed the piston rod to a baffle through a force sensor. Pham et al. (2020), Nguyen and Pham (2021a, 2021b) also developed a friction test bench based on an electric cylinder, which can simulate atmospheric temperature and humidity conditions. They set the force sensor connected to the front of the piston rod outside the environmental simulation room. Figure 4 shows several test setups in which the force transducer is fixedly installed.

The force measuring elements in the test setups mentioned above are fixed, but many researchers have also installed the force measuring elements in a mobile manner. In this manner, the cylinder barrels of the pneumatic cylinders under test are fixed and the two ends of the force measuring elements are connected to the driving elements and the pistons of the pneumatic cylinders. Chen et al. (2010) built a friction test setup for identifying the parameters of the LuGre model and studying the influence of pressure conditions on each parameter. To investigate the influence of vacuums on friction in the cylinder, Tadic et al. (2019) established a friction measurement system where a screw nut driven by a stepper motor was used as the driving element, and designed a negative pressure control system. Azzi et al. (2019) used an electric actuator to realise the linear motion of a force sensor and the piston rod of a pneumatic cylinder for studying the influence of the structural shape of the seal on friction. Lin et al. (2021) developed a friction test bench based on the mobile installation of the force sensor, and revealed the influence law of the geometric errors of the inner wall of the pneumatic cylinder on the friction of the seal. Qian et al. (2022d) proposed a novel dual-cylinder friction test platform, in which two cylinders with the same diameter were arranged face-to-face, and the two piston rods were connected through a force sensor. In this way, the influence of the force of compressed air acting on the piston on the friction measurement results was effectively removed. Figure 5 shows several test setups in which the force transducer moves with the piston rod.

Figure 4 (a) test setup built in Belforte et al. (2013) (b) test setup built in Dilixiati et al. (2016) (c) test setup built in Wakasawa et al. (2019) (d) test setup built in Pham et al. (2020), Nguyen and Pham (2021a, 2021b)

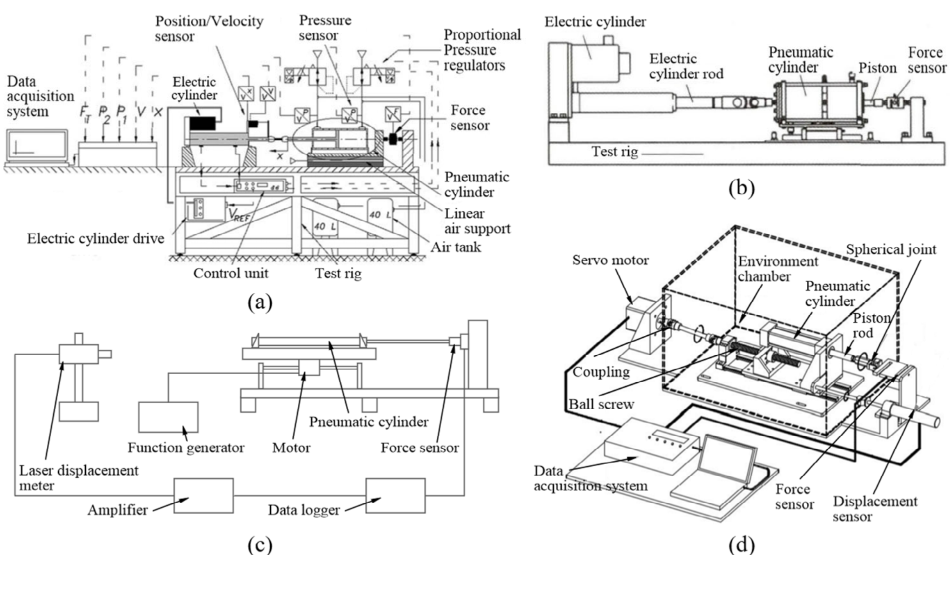


Figure 5 (a) test setup built in Chen et al. (2010) (b) test setup built in Azzi et al. (2019) (c) test setup built in Qian et al. (2022d)

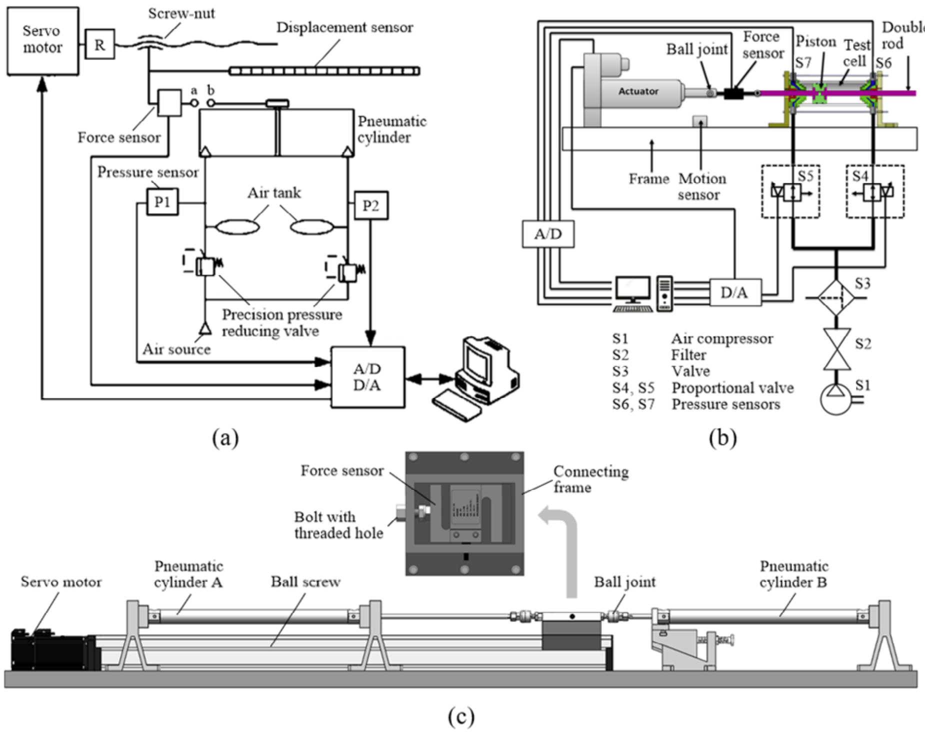
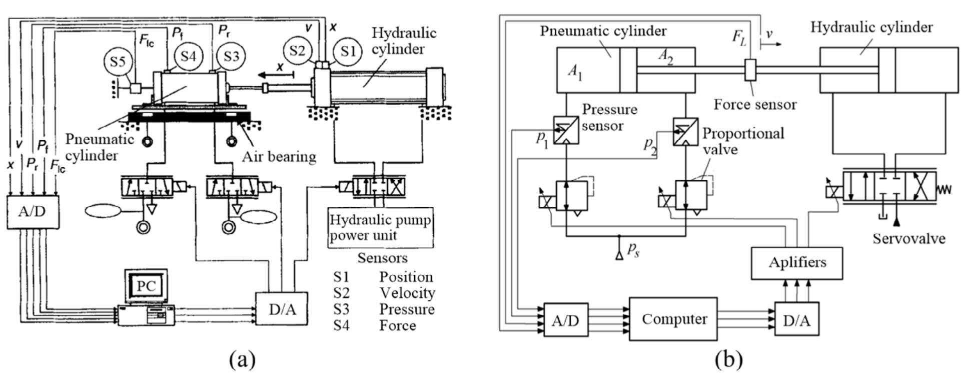


Figure 6 (a) test setup built in Belforteet al. (2003) (b) test setup built in Tran and Yanada (2013)

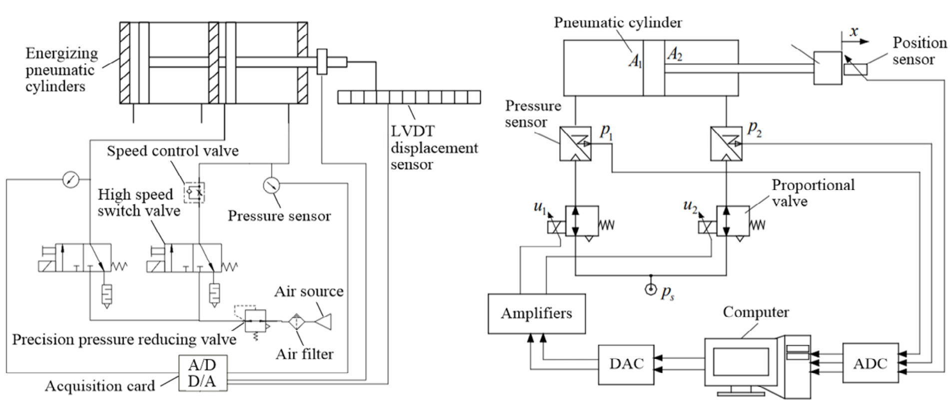


Unlike mechanical structures such as rack and pinion gears and screw nuts that transform motor rotation into the desired linear motion, some researchers have used hydraulic cylinders or pneumatic cylinders as driving elements. Belforte et al. (2003) built a friction test setup using a hydraulic position servo system to drive the piston rod of a pneumatic cylinder at a set position and speed, where the force sensor and the cylinder barrel were fixed. Tran and Yanada (2013) also used a hydraulic cylinder to provide a constant driving speed for the pneumatic cylinder under test, but the difference in the setup was that the force sensor was set between the piston rod of the cylinder and the piston rod of the hydraulic cylinder. Chang et al. (2011, 2012) used a pneumatic cylinder controlled by pressure regulating valves and throttle valves to drive the movement of the cylinder barrel of the pneumatic cylinder under test, and installed the force sensor fixedly. Figure 6 shows the test setups with a hydraulic cylinder as the driving element.

2.3.2 Indirect measurement method

In the indirect method, the friction force of pneumatic cylinders is indirectly calculated by combining Newton's equations of motion, the displacement information of the pistons and the pressure information of the chambers. The indirect measurement method realises the movement of the pistons by controlling the air pressure in the chambers. Zhan et al. (2014) investigated the friction characteristics of energising pneumatic cylinders through two high-speed switching valves, a speed regulating valve, two pressure sensors, and a displacement sensor, and concluded that the friction model of the cylinders can be considered as a superposition of the friction model of each model. Zhang et al. (2015) tested the friction force of a pneumatic cushion cylinder in the cushion phase by using a reversing valve, two speed regulating valves, three pressure sensors and a displacement sensor supporting speed output, and gave a model function for the cushion phase. Wakasawa et al. (2014) built an experimental setup based on two proportional flow valves, two pressure sensors, a displacement sensor, and three acceleration sensors, and discussed the relationship between friction characteristics and vibration characteristics. Tran et al. (2015, 2017) studied the friction behaviours of the piston of pneumatic cylinders in the pre-sliding phase with the help of two proportional pressure valves, two pressure sensors and a displacement sensor, and gave a new model that can better predict the friction characteristics of the cylinders in the pre-sliding phase. Dagdelen et al. (2019) identified the parameters of the Stribeck friction model through a reversing valve, two flow control valves, two pressure sensors and a displacement sensor. Du et al. (2019) designed a bridge pneumatic circuit based on four switching valves and built a friction test bench based on two pressure sensors and a displacement sensor to prove that the proposed composite friction model can adapt to pneumatic cylinders with different working conditions. Figure 7 shows the test setups using indirect measurement methods.

Figure 7 (a) test setup built in Zhan et al. (2014) (b) test setup built in Tran et al. (2015, 2017)



(a) (b)

For the various friction test setups introduced above, we can summarise:

1 In the indirect measurement method, the friction characteristics can be directly studied by pneumatic servo systems. The indirect measurement method often requires differentiation of displacement or velocity to obtain acceleration information, which makes the acquired acceleration easily drowned in noise, resulting in larger error. Therefore, the indirect measurement method is not suitable for low-friction applications.

2 In the direct measurement method, external linear driving elements can provide a stable motion speed for pneumatic cylinders, and the speed can be adjusted in a wide range. However, the direct measurement method requires an additional friction test platform to be built.

3 The driving elements in the direct measurement method can be roughly divided into: linear modules driven by motors, hydraulic cylinders and pneumatic cylinders. The linear modules can maintain smooth operation at very low speeds, while the hydraulic cylinders have creeping problems at low speeds. Since the compression ratio of air is much larger than the compression ratio of hydraulic oil, the creeping phenomenon of pneumatic cylinders is more serious. Therefore, hydraulic cylinders or pneumatic cylinders should not be used as driving elements in the low-speed tests.

4 In the direct measurement method, the installation forms of the force sensors can be mobile or fixed. The force measurement is sensitive to inertial effects when the force sensors are movably installed, while the force measurement is not affected by inertial forces when the force sensors are fixedly installed.

5 In the direct measurement method, the cylinder barrel or piston rod of the pneumatic cylinder under test can be used as the driven part. In the case where the barrel cylinder is driven, it is difficult to eliminate the problem of misalignments between the piston rod and the cylinder barrel during movement. In the case where the piston rod is driven, the piston rod can be guided to move reciprocally for a few cycles to achieve automatic alignment before the cylinder barrel is fixed.

In general, the direct measurement method has more advantages than the indirect measurement method, and the test results of the direct measurement method are more accurate. Although researchers have done a lot of work on the friction in the pneumatic cylinders, they have never found a model that can take into account various factors and can perfectly characterise the friction in pneumatic cylinders due to the uncertainty of friction. Therefore, the exploration of the friction model of pneumatic cylinders has basically reached a bottleneck. However, there is still room for further advancement in terms of high-precision testing of friction in pneumatic cylinders. For example, the friction test method with two cylinders placed face-to-face based on a frictionless pneumatic cylinder mentioned by Qian et al. (2021) in the Chinese invention patent ‘a friction test device and method for pneumatic cylinders’ is an alternative method with higher accuracy in theory.

3 Adopting advanced control strategies

Scholars at home and abroad have done a lot of work in the research of motion control (from classical control, modern control to intelligent control) of pneumatic cylinders, and the widely used control strategies can be mainly divided into two groups: improved linear control and nonlinear robust control. In terms of improved linear control, scholars mostly use gain scheduling, optimal control, linear robust control, artificial intelligence and other means to make up for the shortcomings of traditional PID, state feedback and other linear control strategies, but the improvement of accuracy is limited. In terms of nonlinear robust control, the theory of control algorithms is more rigorous, and the robust stability and robustness of closed-loop systems have been extensively proven through theory and experiments. Comparing the two, it seems that the latter can generally achieve the higher control accuracy. In addition, some researchers have tried fuzzy control (Shih and Ma, 1998; Bai and Li, 2008; Huang and Shieh, 2009; Qiu et al., 2013), neural network control (Song et al., 1997; Zhu et al., 2001), fuzzy neural network control (Kong, 2007; Liu, 2008), model reference adaptive control (Tanaka et al., 1996), self-correcting control (Richardson et al., 2001), model predictive control (Bone et al., 2015), etc. Although some results have been achieved, they are not ideal. Therefore, this section focuses on the summaries of the previous studies on nonlinear robust control.

Since a large number of assumptions and simplifications are made in the modelling of the motion servo control systems for pneumatic cylinders, the established mathematical models will have un-modelled dynamics and parameter uncertainties. These problems need to be explicitly addressed in the controller design. In general, robust control and adaptive control are the common methods to solve the problems in modelling.

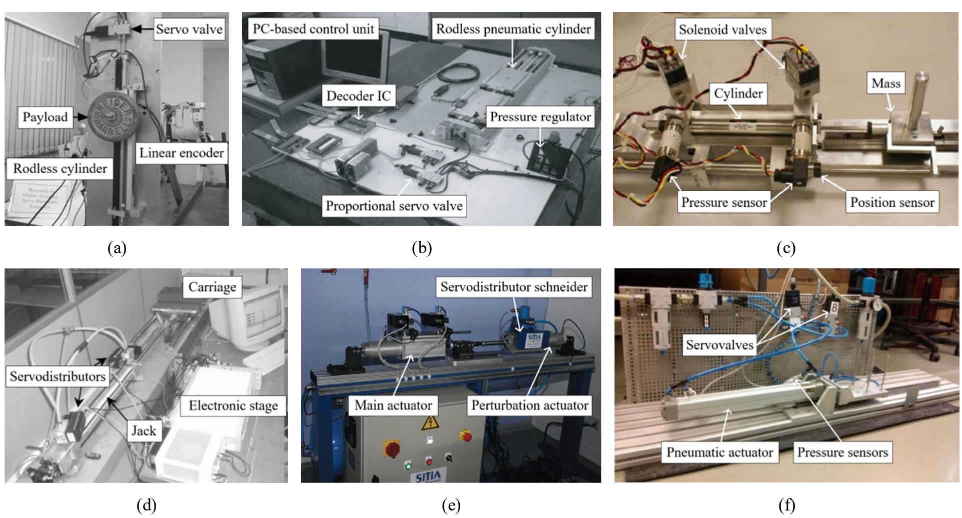
3.1 Sliding mode control

Sliding mode control (SMC) is a simple way to realise robust control. There are many studies on the application of SMC in motion servo control of pneumatic cylinders. Drakunov et al. (1997) established a highly nonlinear fourth-order state space model for a rodless pneumatic actuator, and proposed a SMC method to solve the control challenges of the pneumatic actuation system with Coulomb friction and viscous friction. Song and Ishida (1997) proposed a SMC scheme with strong robustness based on the uncertainty bounds of the structural characteristics of the pneumatic servo system and Lyapunov stability theory, which can only be applied to second-order pneumatic servo systems. Pandian et al. (1997) used differential pressure feedback instead of acceleration feedback, and established a mathematical model of a third-order pneumatic system with displacement, velocity, and differential pressure as state variables. On this basis, they proposed a practical SMC controller with strong robustness to the changes in loads and supply pressures. Surgenor and Vaughan (1997) designed a continuous SMC controller that is more robust than traditional proportional-differential feedback control and proportional-velocity-acceleration control. The continuous SMC controller exhibited better performance than the other two methods over the load mass range of 2.2 kg to 25 kg. Righettini and Giberti (2002) designed a sliding-mode-based nonlinear controller for a single-rod pneumatic cylinder system controlled by two electro-pneumatic proportional valves (flow type), which relies only on a linear encoder. The results showed the designed SMC controller enabled the controlled cylinder to robustly track the desired trajectory over a wide range of speed and load variations. Bone and Ning (2007) designed three control algorithms for the servo positioning problems of a pneumatic system: a position-velocity-acceleration feedback control algorithm with feedforward and deadband compensation, a SMC algorithm based on a linearised object model and a SMC algorithm based on a nonlinear object model. Through experimental comparison, it was found that the nonlinear model-based SMC algorithm had the best performance, and the maximum tracking error was 0.5 mm when tracking the trajectory of $x(t) = 70 \sin 0.5\pi t$. To cope with a series of problems such as nonlinearity, time variation and chatter in the pneumatic servo control system, Lee and Li (2012) proposed a new robust adaptive SMC method (HWB-ASMC+H∞) by incorporating the H∞ tracking technique into the adaptive SMC method developed based on an orthogonal Haar wavelet. The results showed the maximum error of the system was 1.2 mm and 1.8 mm when tracking the trajectory of $x(t) = 100 \sin(0.5\pi t) + 150$ mm at a payload weight of 6 kg and 13 kg, respectively. Barth et al. (2002), Shen et al., 2006) used the averaging method to transform the original discontinuous model into a continuous model for the system in which two solenoid valves were used to control a pneumatic actuator, and then designed a pneumatic adaptive SMC servo controller based on PWM-controlled solenoid valves by virtue of the continuous model. The results showed the adopted means could realise the motion trajectory tracking control of the pneumatic actuator through the solenoid valves, except that the tracking accuracy was poor, which was about 10% of the amplitude for a 0.25 Hz sinusoidal trajectory. Nguyen et al. (2007) used four solenoid valves without pulse-width modulation to control two chambers of a pneumatic cylinder, and designed a model-free SMC method to control the cylinder motion by constructing three operation modes including an energy-saving mode. The results showed the maximum tracking error for a sinusoidal trajectory with an amplitude of 20 mm and a frequency of 0.5 Hz is more than 10% of the amplitude. Hodgson et al. (2012) developed a seven-mode SMC controller that requires pressure information and position information of pneumatic cylinders by adding four additional operation modes based on (Nguyen et al., 2007). The results showed the proposed controller had better steady-state tracking accuracy and fewer switching times of valves than the 3-mode controller. Smaoui et al. (2004) initially configured two three-way proportional directional valves into one five-way proportional directional valve to control two chambers of pneumatic cylinders, and designed a second-order SMC controller based on the super-twisting algorithm, which can not only retain the same robustness as the first-order SMC controller, but also avoid the undesired flutter to a certain extent. Later, they proposed a combined control approach that uses a first-order SMC controller to control the chamber pressure and a second-order SMC controller to control the piston position (Smaoui et al., 2005a). The results showed the maximum position tracking error was close to 3 mm and the maximum pressure tracking error was close to 0.06 bar. In addition, to further reduce the flutter and improve the position tracking performance, they proposed a robust differentiator-controller designed based on a three-order SMC (Smaoui et al., 2005b), in which the differentiator was used to estimate the acceleration signal of the piston to attenuate the noise caused by the differentiation of the velocity signal. Girin et al. (2006) argued that for nonlinear systems, nonlinear control strategies can achieve higher control accuracy, and only the displacement of the piston and the pressure of one chamber need to be measured through the corresponding sensors, while other state variables can be estimated by designing observers. For this purpose, two state observers (high-gain state observer and sliding mode state observer) were designed and their performance was compared. To address the high accuracy and bandwidth requirements of electro-pneumatic systems in aerospace applications, Girin et al. (2009) designed and compared a linear controller based on gain scheduling feedback and a higher order SMC controller. The results showed the latter could meet the basic objectives, but the system bandwidth was limited by the saturation of the control valve and the higher order SMC controller consumed more compressed air. Taleb et al. (2013) and Shtessel et al. (2012) proposed an adaptive second-order SMC algorithm and a Lyapunov-based super-twisting adaptive SMC algorithm, respectively, both of which possess the dynamically adaptive control gains. Relevant experiments based on the setup designed by Girin et al. (2006, 2009) were conducted to confirm that the proposed control algorithms achieved better performance and accuracy of the system. Plestan et al. (2013) proposed a novel SMC algorithm with gain adaptation for a class of nonlinear MIMO (Multiple-Input Multiple-Output) systems with uncertainties and unknown but bounded disturbances, which ensures that the control gain is not overestimated to reduce flutter, and is well used in position-pressure control of electro-pneumatic actuators. Carneiro and Almeida (2015) developed a new control law that includes a robust motion control loop based on integral sliding mode and a pneumatic force control loop based on nonlinear state feedback for pneumatic servo systems. The controller could cope with about five times the payload variation, and could maintain good tracking and positioning accuracy even without a friction model. Deng et al. (2016) designed an adaptive SMC controller incorporating function approximation approach for a diaphragm pneumatic actuator. The results showed the controller was robust to parameter uncertainties and external disturbances, and can cope with large variations in loads. Ren et al. (2019) proposed a fractional order SMC controller to realise the tracking control of the pneumatic servo system, which combines the exponential reaching law. The results show that the maximum tracking error achieved by the fractional order SMC is 1.1 mm, which is 56% lower than the integer order SMC used for comparison. Figure 8 shows the experimental setups of SMC for pneumatic servo systems.

3.2 Robust control based on backstepping method

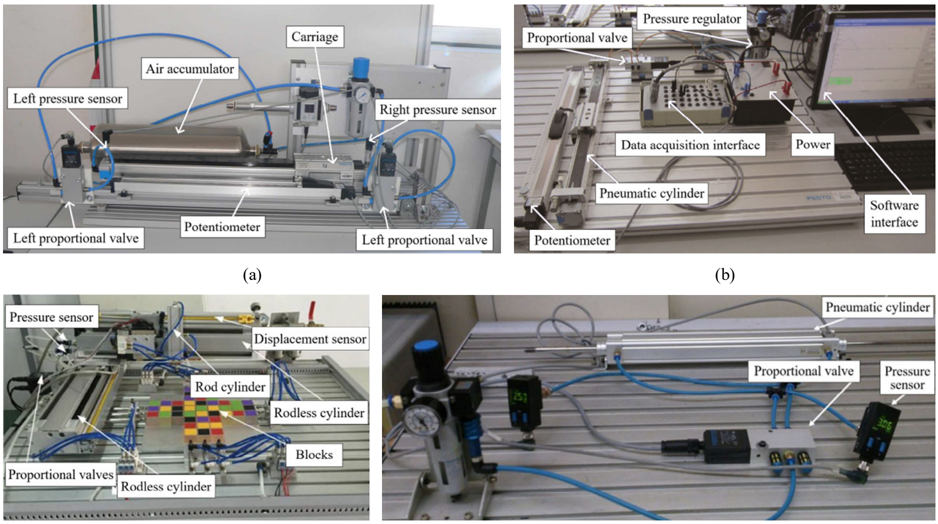
In recent years, some studies have used the backstepping method to design nonlinear robust controllers, which can obtain good tracking results. Smaoui et al. (2006) designed a multivariable backstepping controller and a multivariable SMC controller for nonlinear electro-pneumatic systems, and illustrated experimentally that the backstepping controller did not show significant flutter and could achieve higher positioning accuracy (maximum positioning error of about 1.25 mm, which is about 0.5% of the total displacement value) compared to the SMC controller. Rao and Bone (2008) used the backstepping method to design a multiple-input and single-output nonlinear position control law for a pneumatic system, in which the pneumatic cylinder was controlled by four low-cost two-way proportional valves. The maximum error in tracking a sinusoidal trajectory with an amplitude of 7.5 mm and a frequency of 1 Hz was about 5.5% of the amplitude. Schindele and Aschemann (2009) designed an observer based on the LuGre model to cope with the friction problem at the seals in pneumatic cylinders, and proposed an adaptive backstepping control scheme with a cascade structure that is strongly robust to load variations. The maximum steady-state error achievable in the tracking tests was less than 1 mm. Ren and Fan (2016) designed an adaptive SMC controller based on the backstepping technique for strongly nonlinear pneumatic systems, which does not require dynamic model parameters and uncertain parameter bounds, nor does it require pressure sensors. The results showed the controller had a decent tracking performance. Zhao et al. (2017) considered a nonlinear backstepping controller to improve the response velocity of a pneumatic servo system and a nonlinear error feedback controller to ensure high positioning accuracy. To this end, they proposed a multi-controller strategy incorporating both of the above, and designed an extended state observer to estimate the total nonlinearity of the model. The results showed the controller had very good positioning control performance (positioning accuracy of 0.05 mm and response time of 0.5 s) and slightly poor trajectory tracking performance (some phase lag exists). Soleyman et al. (2017) designed a SMC controller with a 'multi-sliding surface' for the trajectory tracking of a pneumatic servo system by means of a similar backstepping method, in which the motion speed of the pneumatic cylinder was estimated through a designed high-gain observer. The results showed the designed controller could track the reference trajectory, but the accuracy was relatively poor, and the tracking error for a sinusoidal trajectory with an amplitude of 200 mm and a frequency of 0.5 Hz was about 15% of the amplitude. Figure 9 shows the experimental setups of nonlinear robust control for pneumatic servo systems based on the backstepping method.

Figure 8 (a) pneumatic servo system in Bone and Ning (2007) (b) pneumatic servo system in Lee and Li (2012) (c) pneumatic servo system in Hodgson et al. (2012) (d) pneumatic servo system in Smaoui et al. (2004, 2005a, 2005b) (e) pneumatic servo system in Shtessel et al. (2012) (f) pneumatic servo system in Carneiro and Almeida (2015) (see online version for colours)



It can be seen that the main shortcoming of most of the above studies is that the tracking accuracy is guaranteed by high-gain feedback, which consumes too much control energy. In practice, due to the existence of modelling errors and un-modelled dynamics, the feedback gain cannot be taken very high, and needs to be considered comprehensively after taking into account the tracking accuracy and dynamic performance.

Figure 9 (a) pneumatic servo system in Schindele and Aschemann (2009) (b) pneumatic servo system in Ren and Fan (2016) (c) pneumatic servo system in Zhao et al. (2017) (d) pneumatic servo system in Soleymani et al. (2017) (see online version for colours)



3.3 Adaptive control

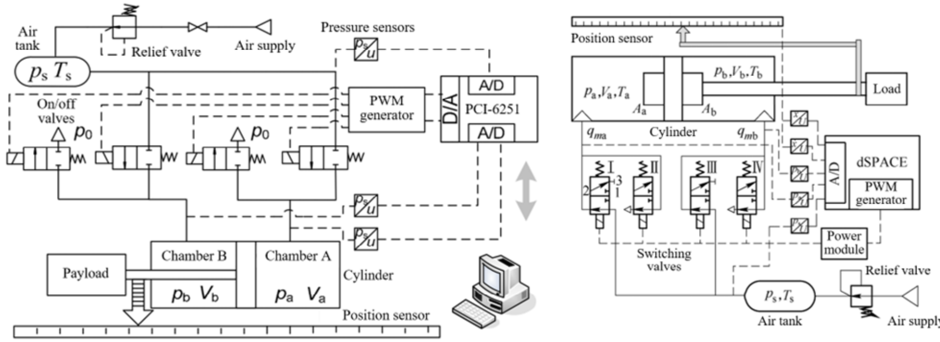
Adaptive control can realise online parameter estimation of nonlinear time-varying pneumatic systems to adapt to changes in the characteristics of the controlled objects. The application of adaptive control in pneumatic position servo systems is relatively much less, and the overall control accuracy is not high (Richardson et al., 2001; Kaasa and Takahashi, 2003). This may be due to the instability of the adaptive control systems caused by un-modelled dynamics or stochastic disturbances in the pneumatic systems. Therefore, it is very difficult to prove the stability of the adaptive control systems through theory. Moreover, the robustness of adaptive control is poor.

3.4 Adaptive robust control

In order to perform high-performance robust control of systems with parameter uncertainties and uncertain nonlinearities, Yao and Tomizuka (1997), Yao (1997) proposed an adaptive robust control (ARC) algorithm by combining adaptive control and robust control. The algorithm can not only reduce the uncertainties of the models through online parameter adaptation, but also provide robust transient performance and steady-state tracking accuracy for closed-loop systems. Its applications in pneumatic position servo systems are quite a lot. Meng et al. (2013a) constructed an ARC controller for achieving the high-precision control of a rodless cylinder controlled by a proportional valve, in which the projection mapping was employed to condition the RLSE (recursive least squares estimation) algorithm to cope with the conflict between the design of SMC and the design of adaptive control. The maximum steady-state error was 2.32% of the amplitude when a 0.5-Hz frequency sinusoidal trajectory was tracked. Then, an integrated direct/indirect ARC was proposed by them (Meng et al., 2013a), which used indirect-type parameter estimation based on a physical model to obtain reliable parameter estimations for unknown models, and adopted a robust control method with dynamically compensated fast adaptation to attenuate the effects of parameter estimation errors, unmodelled dynamics and disturbances. The maximum steady-state error for tracking a 0.5-Hz frequency sinusoidal trajectory was 1.57% of the amplitude. In addition, Meng et al. (20213b) investigated the influence of deadband compensation of proportional directional valves (Meng et al., 2013e, 2018), friction compensation of pneumatic cylinders based on the LuGre model (Meng et al., 2014a), temperature observers for air in the chambers (Meng et al., 2013d) on the control accuracy. Among them, the best result for the maximum steady-state error was up to 1.22% of the amplitude when a sinusoidal reference trajectory of 0.5 Hz was tracked.

For cost reasons, more economical switching solenoid valves were selected to control the pneumatic cylinder action instead of proportional directional valves. For a pneumatic cylinder controlled by four pulse-width modulated switching solenoid valves, Qian et al. (2014) applied recursive backstepping technique to cope with the uncertainty of unmatched model in the system, and constructed a modified direct ARC by combining an adaptive law based on the gradient type of the discontinuous projection mapping and a deterministic robust control strategy. The maximum steady-state tracking error for a sinusoidal reference trajectory of 0.5 Hz is 2.4% of the amplitude. On the basis of earlier studies, Meng et al. (2015) used two sets of high-speed switching valves to control two chambers of a cylinder separately, and designed an ARC controller based on standard projection mapping. The maximum steady-state tracking error for a sinusoidal trajectory of 0.5 Hz is 1.67% of the amplitude. Figure 10 shows the experimental setups of switching solenoid valve-controlled pneumatic cylinder servo systems.

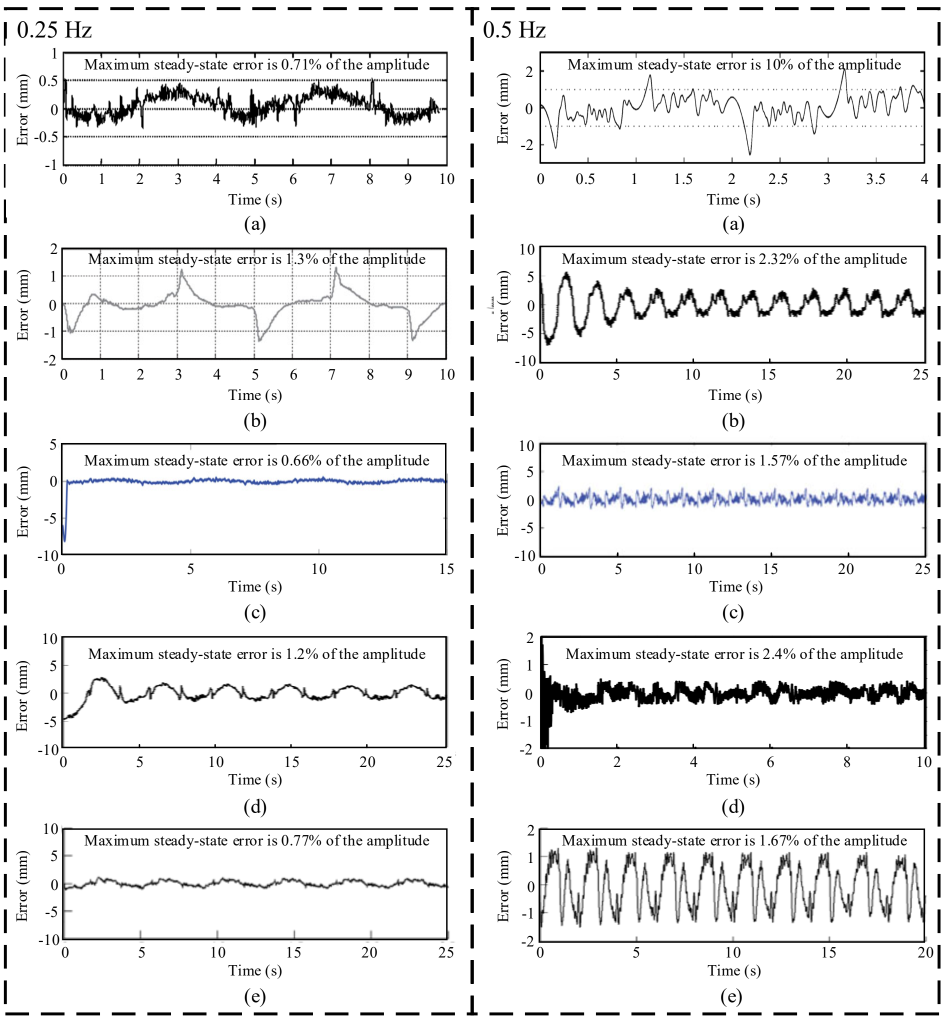
Figure 10 (a) pneumatic servo system in Qian et al. (2014) (b) pneumatic servo system in Meng et al. (2015)



The literature on the application of advanced control strategies to pneumatic servo systems is extensive, most of which focus on tracking control of 0.5 Hz and 0.25 Hz sinusoidal trajectories. Therefore, the test results of these two frequencies are presented in Figure 11.

It can be seen that the ARC algorithm combines the adaptive control algorithm and the robust control algorithm to achieve the complementary advantages of both, and uses projection mapping to design the adaptive control law and the robust control law separately, which effectively solves the conflict between the designs of the two algorithms. Given that the ARC algorithm can effectively cope with the considerable parameter uncertainties and uncertain nonlinearities in the pneumatic position servo systems, the ARC algorithm is a very promising and highly sought-after advanced control algorithm.

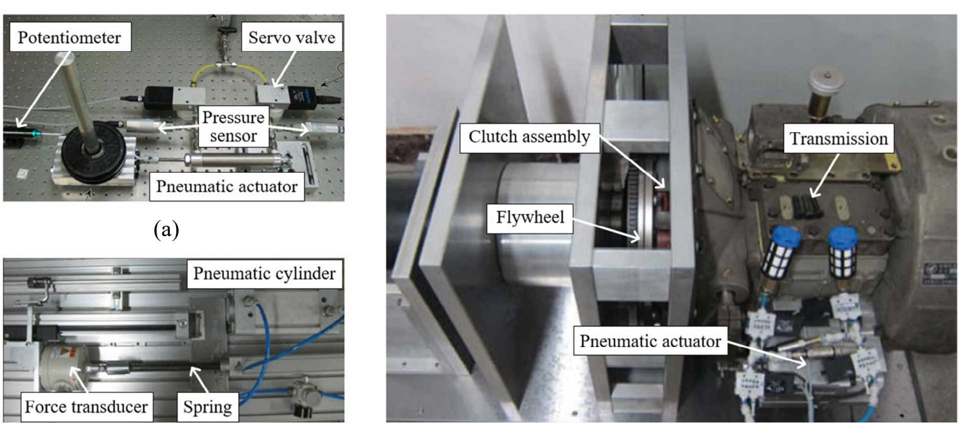
Figure 11 Left (a) experimental results in literature Bone and Ning (2007) (b) experimental results in literature Lee and Li (2012) (c) experimental results in literature Ren et al. (2019) (d) experimental results in literature Meng et l. (2013a) (e) experimental results in literature Meng et al. (2014a) right (a) experimental results in literature Nguyen et al. (2007) (b) experimental results in literature Meng et al. (2013a) (c) experimental results in literature Meng et al. (2013e) (d) experimental results in literature Qian et al. (2014) (e) experimental results in literature Meng et al. (2015) (see online version for colours)



4 Improving pneumatic system stiffness

The valve-controlled pneumatic cylinder systems are elastic systems due to the large compressibility of air. It is well known that the greater the stiffness of elastic systems, the stronger the ability to resist external disturbances. In the traditional trajectory tracking of pneumatic cylinders, it can be observed that the difference between the two-chamber pressure of the pneumatic cylinders and the air supply pressure is relatively large. The stiffness of such valve-controlled pneumatic cylinder systems is not very large, and at least it has room for improvement. Therefore, the potential of the system stiffness is stimulated by means of control, that is the maximum control of stiffness, which in theory can effectively improve the motion control accuracy of the cylinders. There are not many relevant studies in this area. Shen and Goldfarb (2007) used two three-way valves to achieve the simultaneous control of the stiffness and output force of a pneumatic cylinder. Meng et al. (2013c) achieved the adaptive robust output force control of a pneumatic cylinder while maximising or minimising the system air stiffness. Qian et al. (2015) conducted pneumatic cylinder trajectory tracking tests when the two chambers were pressurised and emptied. Through comparing the trajectory tracking errors in the initial stage, it was confirmed that increasing the pressure of the two chambers, that is, increasing the air stiffness, can also effectively improve the motion control accuracy of cylinders. Then, for a valve-controlled cylinder pneumatic system under variable loads, they proposed a composite SMC strategy that can achieve the simultaneous maximum control of motion and stiffness (Qian et al., 2016). In the study, the steady-state peak error was 0.3 mm (1.7% of the amplitude) when the reference trajectory of $x(t) = 17.5 \sin 0.5\pi t + 17.5$ mm was tracked. In addition, they performed the simultaneous maximum control of motion and stiffness for the above system based on a pressure observer (Qian et al., 2017), which had a steady-state peak error of 0.38 mm when tracking the same trajectory above. Figure 12 shows the experimental setups built to improve pneumatic system air stiffness.

Figure 12 (a) pneumatic servo system in Shen and Goldfarb (2007) (b) pneumatic servo system in Meng et al. (2013c) (c) pneumatic servo system in Qian et al. (2015, 2016, 2017) (see online version for colours)



Therefore, improving the pneumatic system stiffness is a good means to improve the motion control accuracy of cylinders, and it is also a direction to continue the research in the future.

5 Improving friction characteristics of pneumatic cylinders

Friction in pneumatic cylinders is mainly generated by the relative movement between the seals and the mating surfaces (cylinder barrel inner wall and piston rod surface). The existence of friction has both advantages and disadvantages. On the one hand, friction has the advantage of increasing damping to make the system stable. On the other hand, friction may cause the wear of sealing rings, the reduction of output force and the creeping phenomenon of low-speed motion. Therefore, the existence of friction in pneumatic cylinders often does more harm than good. The friction characteristics of pneumatic cylinders can be effectively improved by reducing or eliminating the friction force, and this is currently a keen research direction in the field of pneumatics.

5.1 Reducing friction force

Methods to reduce friction in pneumatic cylinders can be roughly classified into three categories: improving sealing forms (sealing rings with coatings, diaphragm seals, special structure sealing rings, etc.), improving lubrication conditions (special lubricants, graphite piston-borosilicate cylinder barrel, gap seals, magnetic fluid seals, etc.) and changing friction state (flutter compensation, etc.).

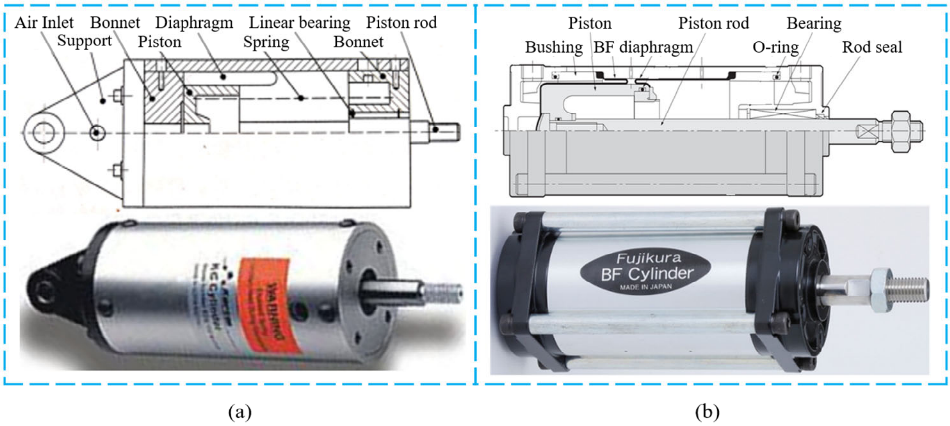
5.1.1 Improving sealing forms

The Swedish company TRELLEBORG has applied coatings to the outer surface of sealing rings, which not only improves the life of the sealing rings, but also effectively reduces the friction in the pneumatic cylinders installed with this type of sealing rings. The KCS series super diaphragm pneumatic cylinders designed by the French company KORTIS use diaphragm capsules for sealing and linear ball bearings to guide the piston rods. The KCS series cylinders have a good sealing performance, a high sensitivity, and a minimum working pressure of 0.01 MPa. However, their stroke is not large due to the limitations of the diaphragm structure, and there is a slight hysteresis during operation. Similarly, there are the rolling diaphragm series cylinders from the American company CONTROLAIR and the SC series linear ball bearing type BF diaphragm cylinders from the Japanese company FUJIKURA. In addition, the CG1□Q, MB□Q, CA2□Q and other series pneumatic cylinders produced by the Japanese company SMC have only a small sliding resistance due to the installation of one-way sealing rings on their pistons, and the sliding resistance increases slightly with the increase of air pressure. The minimum working pressure of these series of cylinders can be as low as 0.01 MPa. Figure 13 shows two series of diaphragm pneumatic cylinders.

In terms of improving sealing forms of pneumatic cylinders, relevant scholars have also conducted some research. Whitney et al. (2014) applied rolling diaphragm technology to the sealing of pneumatic cylinders, making the cylinders have a very good sealing performance and a lower static friction. To address the problem of the short life of V-shaped sealing rings in common low-friction pneumatic cylinders, Belforte et al.

(2014) developed a 'multi-leaf' sealing ring, which ensures a better sealing performance and a minimum contact area with the cylinder barrel inner wall.

Figure 13 (a) KCS series super diaphragm cylinders from KORTIS (b) SC series diaphragm cylinders from Fujikura (see online version for colours)



5.1.2 Improving lubrication conditions

Given that the pneumatic cylinders are mostly elastic sealing structure, pneumatic industries usually use special greases, materials with very low friction coefficient and other means to reduce the friction in common pneumatic cylinders. The low-friction pneumatic cylinders (model with 'S11') developed by the German company FESTO use special two-way sealing rings and special greases, and can move smoothly at a speed of about 5 mm/s in both directions. The Airpel series anti-stiction pneumatic cylinders from the American company AIRPOT are composed of pistons made of graphite material and glass cylinder barrels made of borosilicate material. Their minimum working pressure is less than 0.0014 MPa. In addition, the MQQ and MQP series pneumatic cylinders manufactured by SMC adopt gap seals and built-in ball guide bushes to enhance the lateral load capacity. The minimum motion speed and minimum working pressure of the MQQ series cylinders are 0.3 mm/s and 0.005 MPa, respectively. The MQP series cylinders are not susceptible to eccentric loads due to the use of spherical installation, and can achieve a minimum working pressure of 0.001 MPa and creep-free motion within a stroke of 0.01 mm. Figure 14 shows different series of pneumatic cylinders with improved lubrication conditions.

Chang and Lan (2008) proved that using lubricants with copper oxide and titanium dioxide nanoparticles can effectively reduce the friction force and stick-slip phenomenon of long-stroke pneumatic cylinders through experimental studies. After proposing the PDMS lip sealing technology, De Volder et al. (2007) used hybrid gap-surface tension sealing technology, De Volder et al. (2009a) and hybrid magnetic fluid sealing technology, De Volder et al. (2009b) to achieve the lower friction of miniature hydraulic and pneumatic actuators. The methods mentioned above have effectively improved the friction characteristics of pneumatic cylinders, but they also have their own problems and limitations. Figure 15 shows three low-friction pneumatic cylinders proposed by De Volder et al. (2007).

Figure 14 (a) Airpel series anti-stiction cylinders from Airpot (b) MQ□ series cylinders from SMC (see online version for colours)

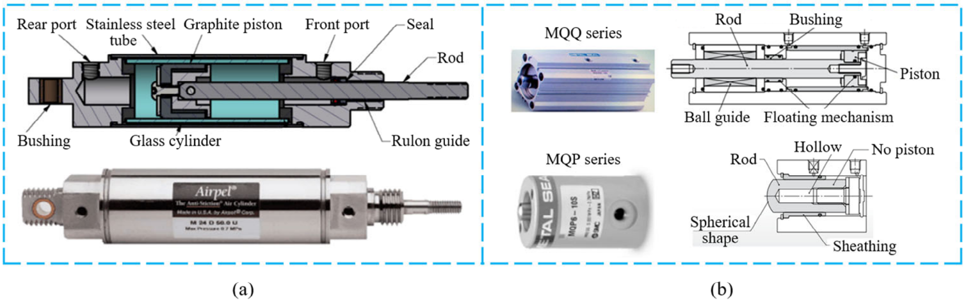
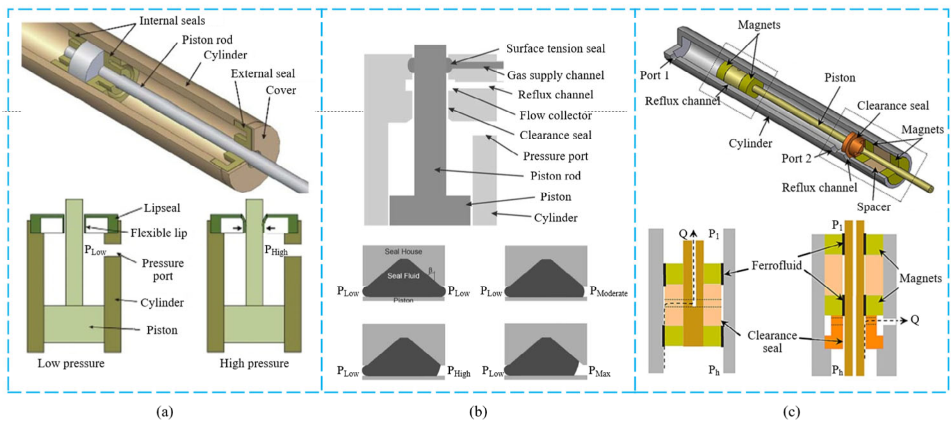


Figure 15 (a) cylinder with PDMS lip sealing (b) cylinder with hybrid gap-surface tension sealing (c) cylinder with hybrid magnetic fluid sealing (see online version for colours)

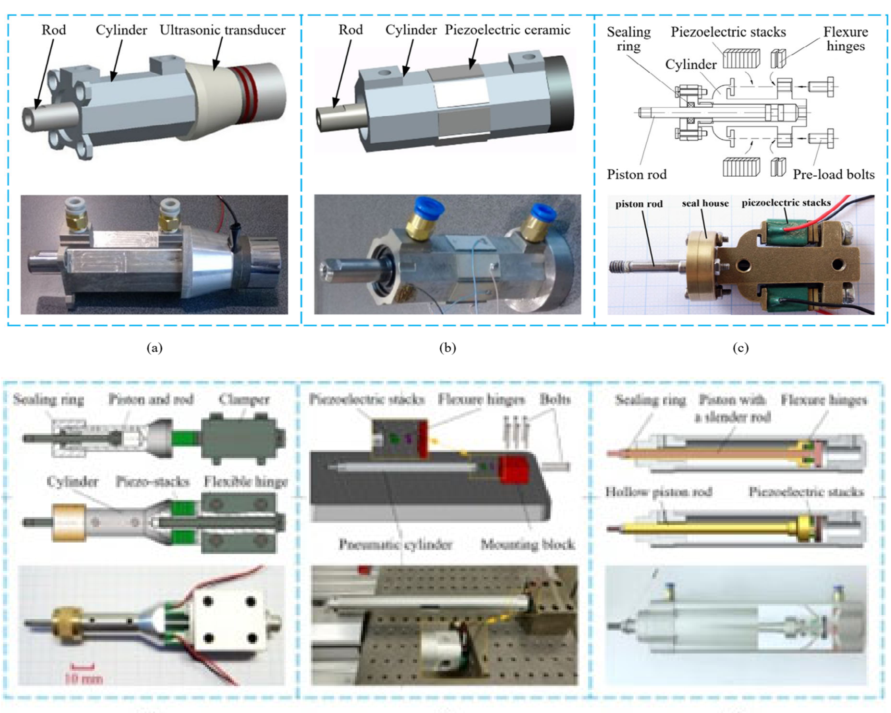


5.1.3 Changing friction state

In recent years, the successful application of ultrasonic friction reduction technology based on ultrasonic vibration (micron-level amplitude above 20 kHz) in machining, driving and mechanical friction reduction has gradually aroused the interest of researchers in the area of pneumatics. Liu et al. (2011) used high-frequency flutter to compensate for the nonlinear motion characteristics of a pneumatic positioning device by installing a piezoelectric actuator between the front of the piston rod and the sliding table. In the research, a PID controller was adopted and 0.1 μm positioning accuracy was achieved. In order to improve the static friction characteristics between rubber and metal, Cheng et al. (2011a), Gao et al. (2015b) built a friction characteristic test bench for planar friction couples of chloroprene rubber-aluminium alloy and nitrile butadiene rubber-aluminium alloy based on ultrasonic vibration. The experimental results showed the existence of the ultrasonic friction reduction phenomenon between rubber and metal. After that, they investigated the effects of vibration frequency, excitation voltage and relative vibration position on static friction characteristics by installing an ultrasonic transducer on the outer wall of a duplex cylinder (Cheng et al., 2011b; Gao et al., 2012).

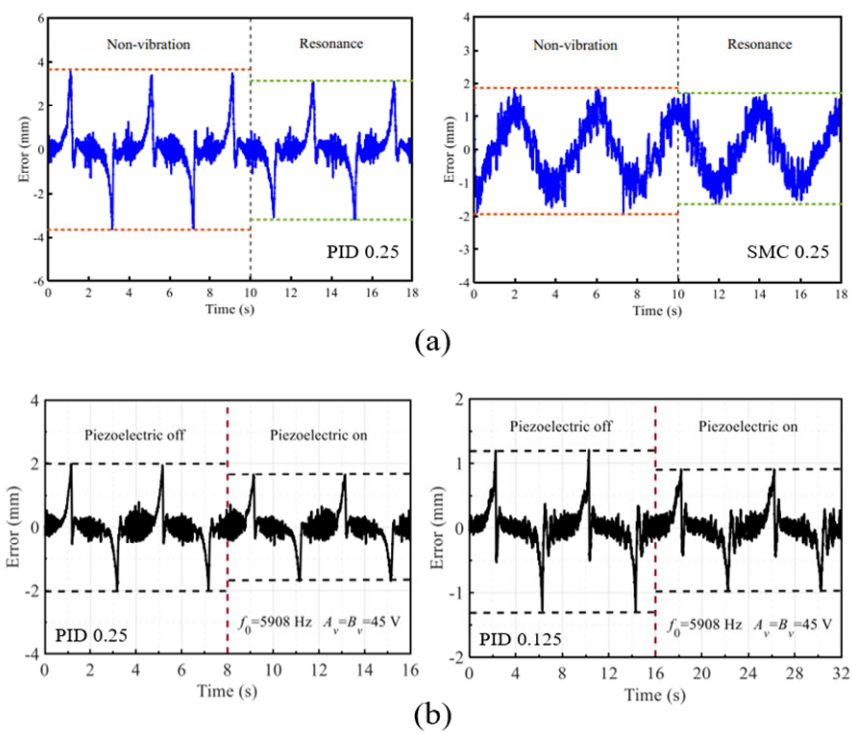
It was concluded that the friction reduction effect was most prominent around the resonant frequency and was enhanced by increasing the excitation voltage within a certain range. Xiao (2013) developed two types of ultrasonic friction reduction pneumatic cylinders based on an ultrasonic transducer and multiple piezoelectric ceramic chips, respectively, which realised the longitudinal vibration mode and bending vibration mode of the cylinder barrels. The results showed the stability and repeatability of the friction force of the pneumatic cylinders subjected to ultrasonic vibration were increased. To study the damping effect of the seal on the ultrasonic transducer, Pham and Twiefel (2015) placed an ultrasonic transducer at the piston of a pneumatic cylinder to cause the axial high-frequency vibration of the piston. They identified the damping and coupling coefficients of the seal by establishing an equivalent circuit model under the contact between the transducer and the seal. Gao et al. (2015a) designed a pneumatic cylinder prototype structure by integrating the piezoelectric stacks into the outer wall of the cylinder barrel. Subsequently, he proposed a vibration friction reduction pneumatic actuator on the basis of the inverse piezoelectric effect, which realised both bending and longitudinal vibration modes (Gao et al., 2016). Through the research of the factors of amplitude, air pressure and the ratio of vibration speed to movement speed, it was found that the bending vibration mode achieved the better friction reduction effect, the rise of air pressure weakened the friction reduction effect, and the friction reduction effect of the longitudinal vibration mode only occurred when the speed ratio was larger than 1.

Figure 16 (a) cylinder proposed in Xiao (2013) (b) cylinder proposed in Xiao (2013) (c) cylinder proposed in Gao et al. (2015a) (d) cylinder proposed in Gao et al. (206) (e) cylinder proposed in Qian et al. (2022b) (f) cylinder proposed in Qian et al. (2022c) (see online version for colours)



The above research mainly focuses on how to maximise the friction reduction of pneumatic cylinders, but there is no detailed research on the impact of the friction reduction on the position servo control accuracy. For this reason, Qian et al. (2022b) proposed a simple test device and method that could realise the longitudinal vibration of a pneumatic cylinder. The test results showed that the high-frequency longitudinal vibration applied to the cylinder could not only reduce the friction, but also improve the motion trajectory tracking accuracy. After that, a novel pneumatic actuator with the piston longitudinal vibration was developed by them, where the static friction decreased by approximately 39% and the dynamic friction decreased by approximately 47% in the case of the longitudinal resonance (Qian et al. (2022c). The preliminary trajectory tracking experiments found that the maximum trajectory tracking errors of the pneumatic actuator could be reduced by about 20%. Figure 16 shows a variety of vibration friction reduction pneumatic cylinders, and Figure 17 shows the results of the effect of frictional improvements on control accuracy in the existing literature.

Figure 17 (a) Trajectory tracking results in (Qian et al. (2022b) (b) trajectory tracking results in (Qian et al. (2022c) (see online version for colours)

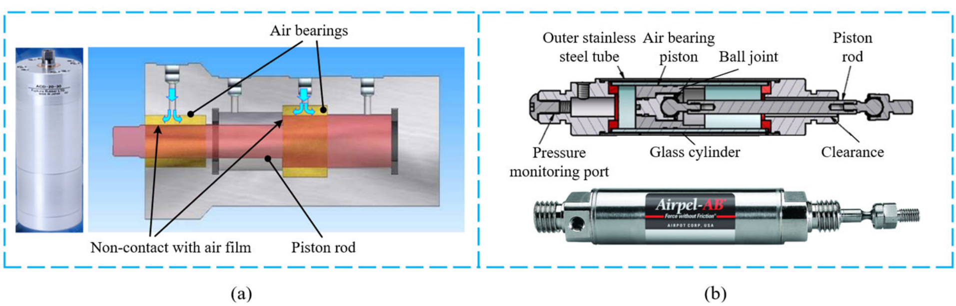


As can be seen, the motion trajectory tracking accuracy of the pneumatic cylinders excited to longitudinal high-frequency resonance was effectively improved. Therefore, making pneumatic cylinders resonate is a feasible way to improve the pneumatic position servo system accuracy, and may be a potential hot research direction in the field of pneumatic technology in the future. It is worth noting that high-frequency vibration friction reduction can only reduce friction to a certain extent, but it cannot completely eliminate the effects of friction.

5.2 Eliminating friction force

With the emergence of the pneumatic cylinders embedded with air bearings, low-friction pneumatic cylinders have realised the transformation to air-floating frictionless pneumatic cylinders. The air-floating frictionless pneumatic cylinders are designed based on the principle of hydrostatic air bearings (Wang et al., 2019). The friction surfaces in these cylinders do not contact each other due to the lubrication of high-pressure air film. Only the negligible friction damping with the air film exists, and thus the frictionless motion is achieved. This is also a form of gap seals. FUJIKURA has launched the AC series air bearing pneumatic cylinders whose core component is porous air bearings sintered from metal. External compressed air enters the gap between the inner wall of the air bearing and the piston-rod assembly through the throttle holes to form an air film, thereby achieving the reciprocating motion with no friction and hysteresis. Their minimum working pressure is 0.01 MPa. AIRPOT has developed the Airpel-AB series air bearing cylinders whose pistons are designed based on the principle of air bearings. The cylinder barrels of these cylinders are made of borosilicate glass, and the pistons do not touch the cylinder barrels during operation to achieve zero friction. Their working pressure must be greater than 0.035 MPa. Figure 18 shows two series of air bearing pneumatic cylinders.

Figure 18 (a) AC series air bearing cylinders from FUJIKURA (b) Airpel-AB series air bearing cylinders from AIRPOT (see online version for colours)



In addition, scholars at home and abroad have also made a great deal of work in the research of frictionless pneumatic cylinders. Corteville et al. (2005) developed a double-rod pneumatic cylinder with light weighting, low friction, and high compliance based on air bearing technology, which was used as an actuator of a humanoid robotic arm. To simulate a zero-gravity environment, Lu et al. (2009) developed an air-suspending frictionless cylinder with friction force of about 0.005 N. Sun et al. (2015) studied a suspension system based on an air-floating frictionless pneumatic cylinder, and analysed the influence of the annular pressure relief groove at the tail of the air-floating piston on the bearing capacity and stability through simulations. Zhu (2016) proposed a frictionless pneumatic cylinder with two check valves and a floating connection mechanism, and constructed a high-precision pneumatic load system. Liu and Zhao (2018) optimised the structural dimensions of the core component (air-floating piston) of an air-floating frictionless pneumatic cylinder by using the basic particle swarm optimisation algorithm, and they constructed a pneumatic gravity compensation system based on the designed air-floating cylinder (Liu et al., 2018, 2020). Qian et al. (2022a) conducted simulation optimisations and working condition analyses of the structure of the proposed new double-acting air-floating pneumatic cylinder to guide the selection of its core parameters.

In summary, the frictionless pneumatic cylinders based on air-suspending technology can almost minimise the friction and are more convenient to achieve, so it is an important development direction of low-friction pneumatic cylinders. However, due to the extremely low friction damping of the frictionless pneumatic cylinders, it is theoretically easy to cause the system to oscillate, which makes it difficult for the frictionless pneumatic cylinders to achieve stable motion control.

6 Conclusions and prospect

The main reasons for the unstable motion of pneumatic cylinders is the existence of friction and the compressibility of air. Researchers often think of achieving high-precision control of pneumatic cylinders through friction compensation. As we all know, friction has many disadvantages such as uncertainty and time-variant. Therefore, the actual friction model of pneumatic cylinders is difficult to accurately establish. Advanced control strategies can improve pneumatic cylinder motion control accuracy, but the results available show that the improvement is limited and has almost reached a bottleneck. Increasing the air stiffness of the system is an effective means, but little research has been done in this area. Finally, in order to improve the pneumatic cylinder control accuracy, researchers have also made many contributions to the improvement of pneumatic cylinder friction characteristics, but only a small amount of literature has investigated the impact of friction improvement on control accuracy. In our future research, the overall elimination of friction in pneumatic cylinders will be considered, so that the uncertainty parts of it can be deleted. However, friction is not useless, and the damping contained in it can help promote system stability. If there is no friction in the pneumatic systems, the system may oscillate. Therefore, the introduction of a constant or controllable damping can make the system reach the equilibrium position without oscillation and quickly. A pneumatic system that can achieve such effect will be a very potential direction in the development of high-precision pneumatic servo technology in the future.

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