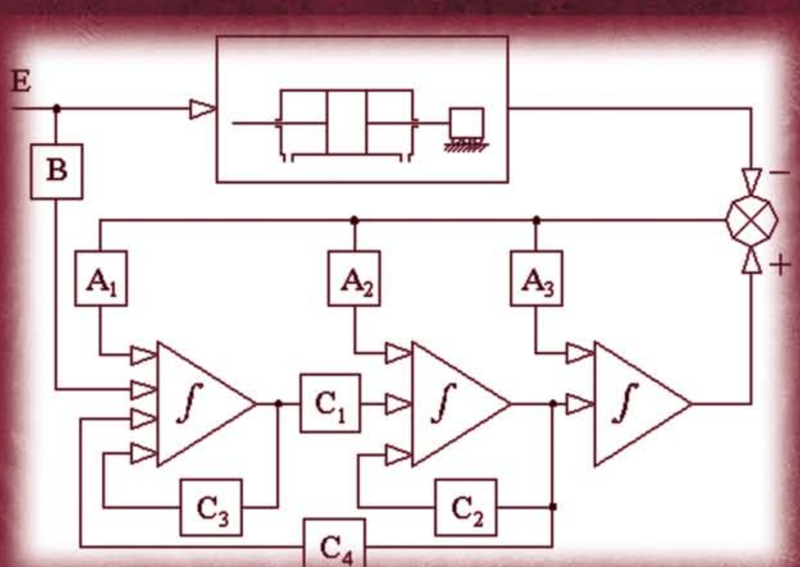
Pneumatic Actuating Systems for Automatic Equipment

Structure and Design



Igor L. Krivts
German V. Krejnin

CRC Taylor & Francis
Taylor & Francis Group

Pneumatic Actuating Systems for Automatic Equipment

Structure and Design

Pneumatic Actuating Systems for Automatic Equipment

Structure and Design

Igor L. Krivts
German V. Krejnin

CRC Taylor & Francis
Taylor & Francis Group
Boca Raton London New York

A CRC title, part of the Taylor & Francis imprint, a member of the Taylor & Francis Group, the academic division of T&F Informa plc.

CRC Press
Taylor & Francis Group
6000 Broken Sound Parkway NW, Suite 300
Boca Raton, FL 33487-2742

Version Date: 20141006

International Standard Book Number-13: 978-1-4200-0446-5 (eBook - PDF)

Visit the Taylor & Francis Web site at
http://www.taylorandfrancis.com

and the CRC Press Web site at
http://www.crcpress.com

Preface

When design engineers begin to develop some form of automatic equipment, they are confronted with two important problems: the first one is related to the mechanical and control design of a device that has functional property; the second problem is a commercial one that pertains to designing with reference to the cost of manufacture.

In order to solve the first problem, especially when an automatic control of complex motion is required, a wide knowledge of the principles underlying those mechanical movements, which have proved to be successful, is very helpful, even to the design engineer who has had extensive experience.

The second problem mentioned, that of cost, is directly related to the design itself, which should be reduced to the simplest form consistent with successful operating. Simplified designs usually are not only less costly, but more durable. Almost any action or result can be obtained mechanically if there are no restrictions as to the number of parts used and as to manufacturing cost, but it is evident that a design should pass the commercial as well as the purely mechanical test. In this connection it is advisable for the design engineer to study carefully the mechanical movement systems, which actually have been applied to commercial machines.

Currently, electromechanical, hydraulic, and pneumatic drives are most widely used as actuation systems in automation equipment. However, all these actuation systems have serious deficiencies, limiting their inherent performance characteristics.

Electrically driven actuators are normally used where movement is required for a number of intermediate positions, particularity when these positions need to be changed easily. They can also control speed and acceleration rate to a very high accuracy independently of the load. This allows very smooth motion to be performed in situations where this is a critical performance factor. In addition, electromechanical actuators can be used where more complex motion profiles are needed and for advanced motion control functions such as registration, contouring, following and electronic cam generation.

The use of electrical motors without torque-magnifying reducers is limited to direct-drive systems, which must employ large DC torque motors that are heavy and inefficient. To increase the torque output to useful levels, gear reducers are almost universally employed. However, there is an increase in torque-to-weight and power-to-weight ratio must be traded off against the large increase in reflected inertia, which increases with the square of the gear reduction value.

Using conventional rotating electrical motors to achieve linear motion requires transformational elements such as screw (ball screw or ACME screw and nut) or a timing belt. Where precision, thrust, and duty cycle are of paramount importance, ball screw models are frequently the best solution. The ACME screw models, which tend to be lower in cost, are an excellent solution unless the application calls for high thrusts. Screw-drive actuators are limited in speed and stroke by screw critical speed, and are offered in rod and guided-rod configurations for thrusting applications.

Belt drive actuators are available for longer stroke lengths or when higher speeds are required. However, because their belts are usually elastic, the screw drive models are typically more accurate and better suited for applications requiring rapid settling. Belt-drive actuators often require a gear reduction to get the mechanical advantage required by the motor to move the load.

The main advantage of electrical motors with transformational elements is that they allow using a low-cost motor that delivers high torque but runs at low speeds.

Electrical linear motors are used in applications requiring the highest speeds, acceleration, and accuracy. These direct-drive linear motors represent a departure from traditional electromechanical devices. Assemblies such as ball screws, gear trains, belts, and pulleys are all eliminated. As the name implies, the motor and load are directly and rigidly connected, improving simplicity, efficiency, and positioning accuracy. The acceleration available from direct-drive systems is remarkable compared with traditional motor drives that convert rotary motion to linear motion. The performance benefit is also substantial. There is no backlash, and because feedback resolution is high, direct-drive systems can be counted on to deliver superior repeatability and stiff, true positioning. However, direct-drive systems are more sensitive to the actuator's force/torque ripple, and they also suffer from lower continuous force/torque compared to geared actuators. Moreover, they are sensitive to load because of the lack of the attenuation effect of a gearbox.

The primary limitation of electromechanical drives is their relatively low power-to-weight, power-to-volume ratio, and payload-to-weight ratio. Table 0.1 represents these characteristics for electrical, hydraulic, and pneumatic motors. From this table it can be seen that the electrical motor has the poorest ratios, and this limits its application.

Generally, the linear motion systems with electrical motor and transformational elements have positioning accuracy of about 5–10 μm (best case) and velocity of up to 500–600 mm/s (second). For systems with electrical linear motors these parameters are the following: positioning accuracy up to 0.1 μm, and velocity up to 1.5 m/s.

Electrohydraulic servo systems provide positioning accuracy on a par with electromechanical systems, offering considerable force, excellent stiffness, and moderate speeds.

Hydraulic actuators (this actuator type is a direct-drive system), which have the highest torque and power density characteristics of any of the

TABLE 0.1

Characteristics of Motors

	Pneumatic Motor	Hydraulic Motor	Electrical Motor
Power-to-Weight Ratio (kW/kg)	0.3–0.4	0.5–1	0.03–0.1
Power-to-Volume Ratio (kW/m³)	$1 \cdot 10^{-3}$–$1.2 \cdot 10^{-3}$	~$2 \cdot 10^{-3}$	$0.05 \cdot 10^{-3}$–$0.2 \cdot 10^{-3}$
Payload-to-Weight Ratio (N/kg)	11	20	3.5

actuation methods, are capable of performing tasks that involve the application of thousands of Newton-meters of torque and many kilowatts of power output. Other aspects that make a hydraulic actuator useful are the low compressibility of hydraulic fluids and high stiffness which leads to an associated high natural frequency and rapid response. This means that automatic equipment using hydraulic actuators can execute very quick movements with great force. Additionally, the actuators tend to be reliable and mechanically simple as well as having a low noise level and relative safety during operation. As for this method of actuation, design characteristics are well known, so the process of design is made easier.

One of the larger concerns with hydraulic systems is the containment of the fluid within the actuation system. Not only can this cause contamination of the surrounding environment, but leakage can contaminate the oil and possibly lead to damage of interior surfaces. In addition, the hydraulic fluid is flammable and pressurized, so leaks could pose an extreme hazard to equipment and personnel. This adds to undesirable additional maintenance to maintain a clean, sealed system. Other drawbacks include lags in the control of the system due to the transmission lines and oil viscosity changes from temperature changes. In fact, such temperature changes in the fluid can be drastic enough to form vapor bubbles when combined with the changes in fluid pressure in a phenomenon called *cavitation*. During operation, as temperature and pressure fluctuate, these bubbles alternately form and collapse. At times, when a vapor bubble is collapsing, the fluid will strike interior surfaces that have vapor-filled pores and high surge pressures and will be exhibited at the bottom of these pores. The cavitation can dislodge metal particles in the pore area and leave a metallic suspension within the fluid. The degradation of the interior surfaces and contamination of the fluid can result in a marked drop in the performance of the system.

Basically, the hydraulic actuation systems can develop controlled stroke speeds of up to 1 m/s, and positioning accuracy of about 1–5 μm.

Nearly 70% of today's positioning applications move loads of between 1 and 10 kg with accuracy between ±0.02 and ±0.2 mm. Electromechanical and electrohydraulic systems are overdesigned for these requirements. Electropneumatic motion systems have high application potential in this field.

Pneumatic actuators are still among the most widely used in automation equipment. As a rule, these actuators are direct-drive systems, too. Pneumatic actuators have been used in devices when lightweight, small-size systems with relatively high payload-to-weight ratio are needed. These actuators are selected for automation tasks as a preferred medium because they are relatively inexpensive (this technology costs approximately 15 to 20% of an electrical system), simple to install and maintain, offer robust design and operation, are available in a wide range of standard sizes and design alternatives, and offer high cycle rates. In addition, pneumatics is cleaner and nonflammable, making it more desirable in certain environments. Furthermore, pneumatic devices are less sensitive to temperature changes and contamination.

Pneumatic actuators are ideally suited to fixed travel applications and the control of force, where precise control of speed is not a prime requirement. In this case, hard mechanical stops are usually positioned along the length of the actuator. Though this adds a certain amount of adaptability, the stops are not truly programmable. They will need to be moved manually should an alternate position be desired.

New technologies today integrate the power of air with electronic closed-loop control. The combination of these technologies can provide much higher acceleration and deceleration capabilities than either one used alone. This position, velocity and force-control system technology is typically lower in cost compared with electrical motion systems. Such servo pneumatic systems retain the advantages of standard pneumatics and add the opportunity for closed-loop, controlled, programmable positioning to within fractions of a millimeter in systems in which positions can be approached rapidly and without overshoot, and provide stability under variable loads and conditions and adaptive control for optimized positioning. Generally, servo pneumatic actuators are similar to hydraulic servo actuators and use proportional or servo pneumatic valves, relying on the integration of electronic closed-loop controlled servo techniques. However, these actuators have the following major disadvantages: poor damping, high air compressibility, strong nonlinearities, and significant mechanical friction. Now, thanks to advances in pneumatic control theory, the combination of fast-acting valves, advanced electronics, and software, servo pneumatic systems are capable of positioning accuracy on the order of 0.05 mm. That level of precision is sufficient for an estimated 80% of typical industrial positioning requirements.

Generally, the linear motion systems with pneumatic actuators and hard mechanical stops have positioning accuracy about 10 μm (best case) and velocity of up to 2.5 m/s. For systems with servo or proportional valves these parameters are positioning accuracy up to 50 μm, and velocity up to 2.5 m/s.

The development of modern pneumatic actuation systems is to be seen as an evolution in mechatronic systems, when integrated with mechanical and electrical technologies, electronic control systems, and modern control algorithms. Trends in actuator pneumatic development can be broken down into the following areas:

- Development of new actuators (frictionless and flexible units in particular) and specialized actuators
- Optimization of component performance and reliability (with particular attention to miniaturizing valves and actuators, reducing frictional force, and standardization)
- Development of new control algorithms and control units with new interfaces between very low control signals and high power pneumatic signals
- Integration with sensors and control electronics to implement intelligent servo systems

All design problems are compromises. It is often practical to have a few parameters that make the compromise explicit. These parameters are called *design parameters*, and it is very important to define those that allow reaching the maximum efficiency in a fine-tuning process of the design on line of the automatic equipment.

This book describes many of the most-applied pneumatic actuating systems, which can be used in various classes of mechanisms, a study of such mechanical movements is particularly important to the designers and students of designing practice owing to the increasing use of automatic equipment in almost every branch of manufacture. The book discusses not only these actuator embodiment principles, their mathematical models, and methods of parameter calculation; but also included are many practical examples and exercises designed to enhance the reader's understanding of the concepts.

Practically, all the pneumatic actuating systems and their components shown in this treatise have been utilized on automatic machines of various classes.

This book is intended for engineers, system designers, and component manufacturers working in the field of pneumatics used in factory automation.

Feedback on the Book

We look forward to receiving readers' comments and corrections of any errors in this book. We encourage you to provide precise descriptions of any errors you may find. We are also open to suggestions on how to improve the textbook. For this, please e-mail the first author: igor_krayvitz@amat.com.

Acknowledgments

Most often, a book is based on previous experience that is complemented with modern advances, and this is especially true for this book. The majority of material described in this book was prepared during collaboration at the Mechanical Engineering Institute of the Russian Academy of Sciences.

We wish to thank our colleagues, both past and present, at the Mechanical Engineering Institute (IMASH), all of whom have given us help and encouragement in writing this book. We especially want to thank our colleagues in the Department of Actuating Systems; most of these people are mentioned by referencing their work in the bibliography at the end of the book. We would like to particularly acknowledge Doctors E.V. Hertz, V.I. Ivlev, K.S. Solnzewa, V.M. Bozrov, A.R. Taichinov, E.A. Tsuhanova, M.A. Jashina, V.V. Lunev, and S.J. Misurin for their collaboration, support, and help in our research.

For many years we have had effective cooperation with Professor V. Frank and Doctor A. Ulbricht from Dresden University of Technology. We are deeply grateful to them for their inspiring and fruitful work.

We would like to pay tribute to the support and help from staff of the Mechanical Department at Applied Materials (PDC Business Group), especially to Doctor E.Y. Vinnizky for his useful discussions about the configuration and dynamic behavior of pneumatic actuators.

Special thanks must go to Mrs. F. Swimmer for her expertise in the English language, as well as for her support in text preparation.

Authors

Igor Lazar Krivts, Ph.D., is a senior mechanical engineer in the Mechanical Department of the PDC Business Group of the Applied Materials Inc. Prior to his current affiliation, he was a senior researcher in the Mechanical Engineering Institute (IMASH), Russian Academy of Sciences from 1980 through 1991.

Dr. Krivts received his M.Sc. (1975) in mechanical engineering from the Penza Technical University (Russia) and a Ph.D. (1985) in mechanical engineering from the Mechanical Engineering Institute at the Academy of Sciences (Russia).

His current research and engineering interests include pneumatic and hydraulic servo systems and their components, precision mechanisms, robotics, motion systems, and vacuum devices for the semiconductor industry. Dr. Krivts' recent research and development activities involve mechanisms in the areas of instrumentation for semiconductor device manufacturing, especially metrology equipment.

As a result of his wide experience in the development and research field, Dr. Krivts holds 23 patents in collaboration with his colleagues. He has authored or coauthored one book and more than thirty scientific papers published in professional journals and proceedings of scientific conferences.

German Vladimir Krejnin, Ph.D., D.Tech.Sc., is a professor in the Mechanical Engineering Institute (IMASH), Russian Academy of Sciences. He is head of the Department of Actuating Systems and holds the academic secretary position in this institute.

Dr. Krejnin received his M.Sc. (1950) in mechanical engineering from the Bauman Moscow Technical University (Russia). He obtained his Ph.D. (1961) and D.Tech.Sc. (1970) degrees in mechanical engineering, both from the Mechanical Engineering Institute at the Academy of Sciences (Russia). Since 1980 he has been a full professor at the Department of Actuating Systems at Mechanical Engineering Institute (IMASH), Russian Academy of Sciences.

His main research interests lie in the areas of pneumatic and hydraulic servo systems and their components, the dynamics and control of mechanical systems with various types of drives, and the methods of optimal synthesis of such systems.

Dr. Krejnin deals with most branches of mechanical engineering, especially those that involve applications of the actuating systems in different kinds of machines and mechanisms. He has supervised 15 doctoral students, and written 7 books and more than 150 scientific journal articles on various topics in mechanical engineering.

List of Symbols

Subscript "i"indicates the working chamber index, superscript "+" indicates the upstream parameters, and superscript "−" indicates the downstream parameters.

Latin Symbols

A	Effective area of the actuator piston [m^2]
A_D	Effective area of the shock absorber piston [m^2]
A_V	Effective area of the control valve [m^2]
b_V	Viscous friction coefficient for linear motion $\left[\frac{N\cdot s}{m}\right]$
b_ω	Viscous friction coefficient for rotary motion [$N\cdot m\cdot s$]
C_P	Air heat capacity for constant pressure $\left[\frac{J}{kg\cdot K}\right]$
C_v	Flow coefficient [gal/min]
C_V	Air heat capacity for constant volume $\left[\frac{J}{kg\cdot K}\right]$
E	Energy [J]
E_M	Modulus of elasticity (Young's modulus) [Pa]
f_A	Actuator bandwidth [Hz]
F	Force [N]
F_D	Dynamic coulomb friction force [N]
F_{DE}	Desired value of the control force [N]
F_F	Friction force [N]
F_L	External force load [N]
F_M	Electromagnetic force [N]
F_S	Static coulomb friction force [N]
F_{SA}	Shock absorber force [N]
G	Air mass flow [kg/s]
h	Discrete time index
h_S	Specific enthalpy of the air flow [J/kg]
I	Current of the control signal [A]
J	Moment of inertia [$kg\cdot m^2$]
K_A	Acceleration gain $\left[\frac{V\cdot s^2}{m}\right]$
K_D	Derivative gain $\left[\frac{V\cdot s}{m}\right]$
K_P	Proportional gain [V/m]
K_{PA}	Gain of electrical power amplifier
K_{SL}	Slope coefficient of control valve [$1/V$]
K_v	Water flow rate [m^3/h]

K_*	Constant coefficient $\left(K_* = \sqrt{\frac{2 \cdot k \cdot R \cdot T_S}{k-1}} \approx 760 \frac{m}{s}\right)$
k	Adiabatic exponent (for air, $k = 1.4$)
k_S	Stiffness of spring $[N/m]$
k_B	Bellows spring rate $[N/m]$
k_D	Diaphragm spring rate $[N/m]$
L_S	Stroke of pneumatic cylinder $[m]$
L_A	Displacement of the actuator acceleration part (open-loop actuator) $[m]$
L_C	Actuator displacement where it moves with constant velocity (open-loop actuator) $[m]$
M	Torque $[N \cdot m]$
m	Mass of load (moving mass) $[kg]$
m_A	Mass of air $[kg]$
P	Absolute pressure in actuator working chamber $[Pa]$
P_A	Absolute atmospheric pressure ($P_A = 0.1 \cdot 10^6\ Pa$)
P_D	Absolute pressure in working chamber of the shock absorber $[Pa]$
P_S	Absolute supply pressure $[Pa]$
Q_n	Standard nominal flow rate $[m^3/s]$
R	Gas constant $\left(\text{for air, } R = 287 \frac{J}{kg \cdot K}\right)$
s_D	Shock absorber working stroke $[m]$
t	Time $[s]$
t_{CR}	Carrier period $[s]$
t_D	Sampling period $[s]$
t_L	Delay time of the control signal $[s]$
t_M	Mechanical time constant $[s]$
t_P	Pneumatic time constant $[s]$
t_{PA}	Time constant of control valve with power amplifier $[s]$
t_{SM}	Time for actuator starting motion (open-loop actuator) $[s]$
t_V	Switching time of control valve $[s]$
t_*	Time scale factor coefficient $[s]$
T	Temperature of air [K]
U_C	Control signal $[V]$
U_{CRM}	Amplitude of carrier signal $[V]$
U_R	Regulating signal $[V]$
V	Volume of pneumatic chamber $[m^3]$
W^2	Dimensionless inertial load
x	Position of the cylinder piston $[m]$
x_D	Desired value of the control position $[m]$
$\dot{x}$	Velocity of cylinder piston $[m/s]$
$\dot{x}_C$	Constant velocity of steady-state motion $[m/s]$
$\dot{x}_{CS}$	Velocity set point $[m/s]$
$\ddot{x}$	Acceleration of the cylinder piston $[m/s^2]$
y	Displacement of the control valve plug $[m]$

Greek Symbols

α_A	Ratio of the effective areas of the piston actuator ($\alpha_A = A_2/A_1$)
α_R	Ratio of the effective areas of the rod and piston actuator ($\alpha_R = A_R/A_1$)
β	Opening coefficient of the control valve
δ_A	Steady-state positioning accuracy $[m]$

Δ_F	Admissible force error [N]
Δ_R	Admissible position error [m]
μ	Poisson's ratio
ν	Dimensionless viscous friction coefficient
ξ	Actuator dimensionless displacement
ρ	Density [kg/m^3]
ρ_{an}	Density of air under standard conditions (ρ_{an} = 1.293 kg/m^3)
ρ_F	Shock absorber fluid density [kg/m^3]
σ	Pressure ratio
σ_A	Atmospheric and supply pressure ratio ($\sigma_A = P_A/P_S$)
τ	Dimensionless time
$\varphi(*)$	Flow function
φ_*	Value of flow function saturation ($\varphi_* = 0.259$)
Φ	Magnetic flux [Wb]
χ	Dimensionless force load
Ω	Effective area ratio of control valve

Contents

1

Structure of Pneumatic Actuating Systems

The pneumatic actuating servo systems used in automatic devices have two major parts: the power and control subsystems (Figure 1.1).

The main part of the power subsystem is the motor, which may be of the rotating or linear type. Basically, this device converts pneumatic power into useful mechanical work or motion. The linear motion system widely uses the pneumatic cylinder, which has two major configurations: single or double action. For the single-action configuration, the cylinder can exert controllable forces in only one direction and uses a spring to return the piston to the unenergized position. A double-action actuator can be actively controlled in two directions. In the case of rotary actuation, the power unit is a set of vanes attached to a drive shaft and encased in a chamber. Within the chamber, the actuator rotates by differential pressure across the vanes and the action transmits through the drive shaft.

Most often, the pneumatic actuator has the direct-drive structure; that is, the output motor shaft or rod is the actuator output link. However, sometimes the transmission mechanisms are installed after the motor; in this case, the output shaft is the actuator output link (e.g., in the rotating actuator where the pneumatic cylinder is used as the motor).

Actuator state variable sensors are the input elements of the control subsystems. In general, the displacement, velocity, acceleration, force, moment, and pressure can be measured in the pneumatic actuator. Different sensor designs can read incrementally or absolutely; they can contact a sensed object or operate without contact; and they span a broad range of performance and pricing levels. Linear-position sensors are widely used as feedback elements for motion control in pneumatic actuating systems; there are precision linear potentiometers, linear-variable-differential transformers (LVDTs), magnetostrictive sensors, and digital optical or magnetic encoders.

The important part of the control subsystem is the command module (or task controller), which stores the input information (such as desired positioning points, trajectory tracking, velocity, or force value) and selects them via input combinations. For example, in the positioning actuator, the positions can be stored in the command module (as position list records), and move commands can include additional parameters such as velocity and acceleration.

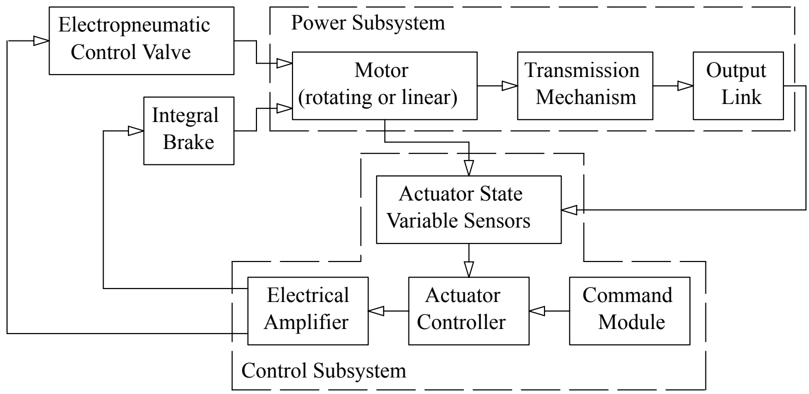


FIGURE 1.1
Block diagram of the pneumatic actuating system.

The central element of the control subsystem is the controller, which provides control, processing, comparing, and diagnostic functions. In general, the controller may be of both types: analog and digital. Currently, more than 90% of all controllers in industry are of the digital type. The main role of this device is to form the control signal according to the control algorithm. The most common form of process controller used industrially is the PID (proportional + integral + derivative) controller. PID control is an effective method in cases where the plant is expressed as a linear model, and the plant parameters do not change with wide or prolonged use. Owing to the compressibility characteristic of the air and high friction force, the pneumatic actuator system is very highly nonlinear, and the system parameters are time variant with changes in the environment. There are main causes, which are limited application of PID control in the pneumatic actuator systems.

For pneumatic actuators, the most common and successful controller is the so-called state controller or PVA (position, velocity, acceleration) controller.[163] In this case, the control signal is a function not only of the positioning signal, but also of the velocity and acceleration signal of the output link motion (for the positioning actuator).

As noted above, in pneumatic actuators, the dynamics of the plant change during performance. In this case, to improve the control performance, an adaptive control system with the controller adjusted bases on the identification results of the plant can be used.

Neural network control and a control algorithm using fuzzy inference are effective for a nonlinear plant. These techniques are applied in the pneumatic actuator controller.[22]

The controller output signals are sent to the electropneumatic control valve via the electrical amplifier. In the pneumatic actuator, the control valve is the interface between the power and control subsystems. This device is a key element in which a small-amplitude, low-power electrical signal is used to provide high response modulation in pneumatic power.

In general, there are three types of electropneumatic control valves — servo, proportional, and solenoid — used in the pneumatic actuator. These valves are available in one-, two-, or three-stage designs. A single stage is a directly operated valve. Two-stage valves consist of a pilot stage and a main stage. Three-stage valves are similar, except that the pilot stage itself is a two-stage valve. Three-stage valves are used in situations where one anticipates very high flow.

The distinction between servo valves and proportional valves is inconsistently defined, but in general, servo valves provide a higher degree of closed-loop control. Traditionally, the term "servo valve" describes valves that use a closed-loop stage spool position back to the pilot stage or drive, either mechanically or electronically.

Proportional valves displace the main-stage spool in proportion to a control signal but normally do not have any means of automatic error correction (feedback) within the valve. Many proportional valves are modified versions of four-way, on/off solenoid valves, in which proportional solenoids replace conventional solenoids. In operation, the solenoid force is balanced by a spring force to position the spool in proportion to the input signal. Removing the centering springs and adding a positioning sensor to the end of the spool can improve the positioning accuracy. The sensor signal then cancels the solenoid signal when the spool reaches the specified position.

Some manufacturers are producing proportional valves that are essentially servo valves made to mass-production specifications, with much greater tolerance allowances and looser fits than in their standard servo line. However, adding electronic feedback results in performance characteristics almost as good as those of a servo valve. In many cases, this results in performance that is perfectly suited to an application at a lower cost.

Solenoid valves are electromechanical devices that use a solenoid to control valve actuation. These devices are a fundamental element of the pneumatics and have high reliability and compact size. Standard models are available in both AC and DC voltages. The solenoid valve is low cost and universal in pneumatic systems operating with on/off control (e.g., it can be an effective solution for repeated stops in two positions). Using the on/off solenoid valve with a PWM (pulse width modulation) control method allows one to achieve the equivalent performance in proportional continually operation of the flow or pressure control. In this case, there is able to replace the solenoid valve instead of the expensive servo or proportional valve.

In some pneumatic positioning and speed control systems, the actuator consists of an integral brake. Usually, a proportional brake is linked to the actuator output link. A programmable controller provides a control signal to the brake and electropneumatic valves based on the stored program. In this case, for one of the possible configurations, the actuator has the on/off solenoid valve, which drives the pneumatic cylinder, and servo function can be achieved via the electric current that is sent to the brake. This type of combined technology system is low in cost, and provides moderate dynamic and accuracy performance.

Usually, pneumatic actuating systems are connected to compressed air lines with pressure from 0.3 to 1 MPa. Air compressors with pump technologies include positive displacement (piston, diaphragm, rotary vane, and screw styles) and nonpositive displacement (centrifugal, axial, and regenerative blowers) and provide air at the necessary pressure. As a rule, the compressor has an integral tank for compressed air storage, a coarse filter, an air dryer, and a pressure regulator.

The removal of moisture from compressed air is important for servo pneumatic systems. Moisture in an air line can create problems that can be potentially hazardous, such as the freezing of control valves. This can occur, for example, if very high-pressure air is throttled to very low pressure at a high flow rate. The Venturi effect of the throttled air produces very low temperatures, which will cause any moisture in the air to freeze into ice. This makes the valve (especially the servo or proportional valve) either very difficult or impossible to operate. Also, droplets of water can cause serious water hammer in an air system, which has high pressure and a high flow rate and can cause corrosion, rust, and dilution of lubricants within the system. For these reasons, air dryers (dehydrator, air purifier, or desiccator) are used to dry the compressed air. Major dryer groupings include refrigerant forced condensation (which removes the water by cooling the air) and desiccants (which adsorb the water in the air with granular material such as activated alumina, silica gel, or molecular sieves). The air can be dried in single or multiple stages.

An additional compressed air service unit is installed on every pneumatic line of the users. A service unit combination usually consists of the following individual units (Figure 1.2): on/off solenoid valve (2), filter (3), pressure regulator (4), and pressure gauge (5). In this case, the system has one input line (1) and two output lines (6) and (8). A lubricator (7) is installed on the output line (8) and supplies lubricant to the pneumatic components. In the

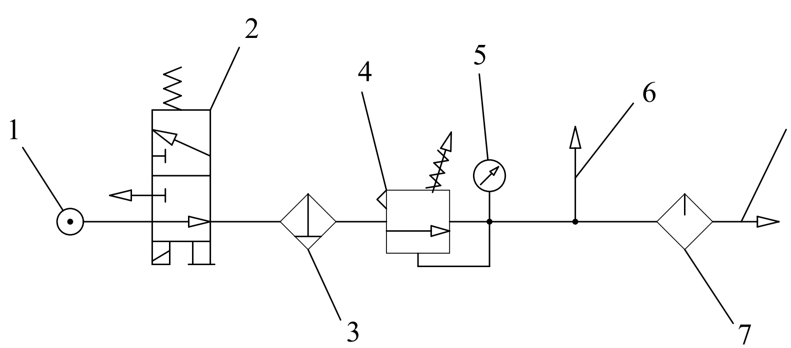


FIGURE 1.2
Block diagram of the compressed air line service unit.

output line (6) the compressed air is clear (without lubrication), which is very important, for example, in a clean-room application in the semiconductor industry. The input line (1) has an on/off control via the solenoid valve (2).

A compressed air filter (3) is used to remove water, oil, oil vapor, dirt, and other contaminants from the compressed-air supply. These contaminants can have a serious effect on the wear and operation of pneumatically operated machinery. In almost all applications, contamination of the air supply could lead to serious performance degradation and increased maintenance costs in terms of actual repairs and production time lost. The proper use and maintenance of compressed-air filters is one sure way to help cut down on these costs. Porous metal and ceramic elements are commonly used in filters that are installed in the compressed-air supply lines. Most pneumatic filters have a removable bowl in which liquids are separated. The condensate that collects in the filter bowl is drained from time to time, as otherwise the air would entrain it.

When selecting a compressed-air filter, it is important to note that the rate particle size of the device is the low end of the size range that is filtered or blocked by the filter. Other important specifications to consider when determining which compressed air filter is best for your system include the standard nominal flow rate or the maximum air volume that will be passed through the filter (generally measured in liters per minute), and the resistance to flow (pressure drop), which is measured in pascals (Pa).

Air-pressure regulators are devices that control the pressure in the air lines of pneumatic tools and machines. These regulators eliminate fluctuations in the air supply and are adjusted to provide consistent pressure. The inlet pressure must always be greater than the working pressure. Usually, the regulator has attached gauges. Just as for the filter, the regulator selection process is very important because its parameters, such as pressure drop, standard nominal flow rate, hysteresis, and transient response, have a significant influence on the dynamics and accuracy of the pneumatic device.

In particular, if positioning servo actuators are required to behave in a large piston stroke range as designed, the supply pressure should be as constant as possible. It is good if the supply pressure variations remain less than 5% of the designed value.

In addition, the extra volume between the pressure regulator and the electropneumatic control valve might improve the system's dynamic behavior. By increasing the value of the extra volume, the dropped pressure can be lowered.[197]

In some applications, where a few drives are operated, two separate supply lines are used: one with high pressure and the other with low pressure. In this case, the drive that moves on the idling mode may be connected to the low-pressure supply line, and the actuator works with high pressure only for the working stroke. This supply system allows for high efficiency.[102]

1.1 Pneumatic Positioning Systems

A pneumatic positioning system has been used widely in robots and manipulators, welding and riveting machines, pick-and-place devices, vehicles, and in many other types of equipment.

Pneumatic positioning actuators can generally be divided into two groups: (1) open-loop and (2) closed-loop position control. Usually, the open-loop pneumatic positioning actuator contains hard mechanical stops. In the simplest case, the system has a pneumatic cylinder, in which two covers play the role of the hard mechanical stops that define the stop positions. Figure 1.3 shows the block diagram of such an actuator. This construction has a piston (1, Figure 1.3) with two cylindrical parts (2 and 3), which are made with two cover cavities (4 and 5) and provide the air-cushioning mechanism. The solenoid control valve (6) connects the pneumatic cylinder to the supply pressure and exhaust port according to the control algorithm. Adjustable throttles (7 and 8) define the maximum value of the piston velocity. The piston (1) includes two permanent magnets (9 and 10) and two proximity sensors (11 and 12) attached to the outside of the cylinder tube. These provide a noncontact indication of cylinder piston position. As the piston approaches,

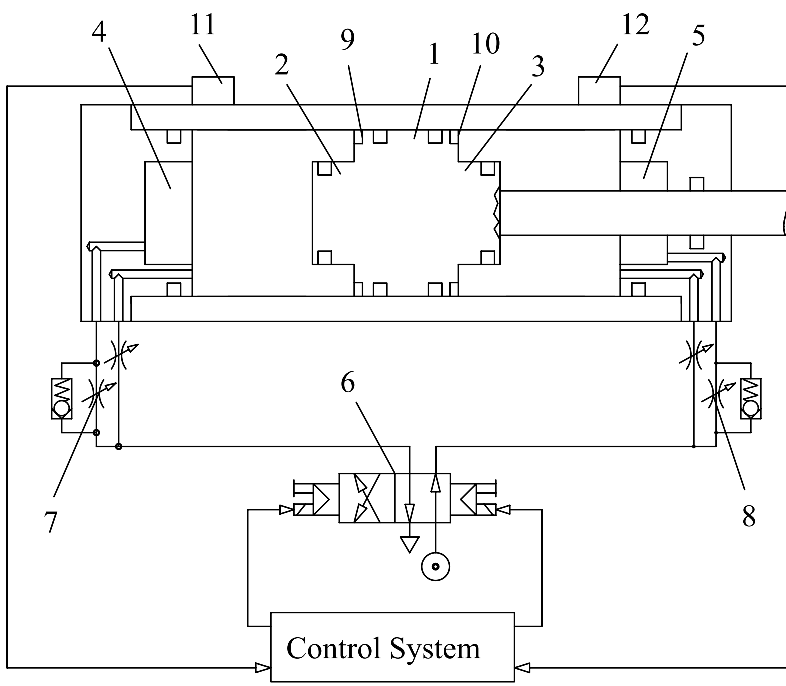


FIGURE 1.3
Block diagram of the pneumatic cylinder with two positioning stops on the ends.

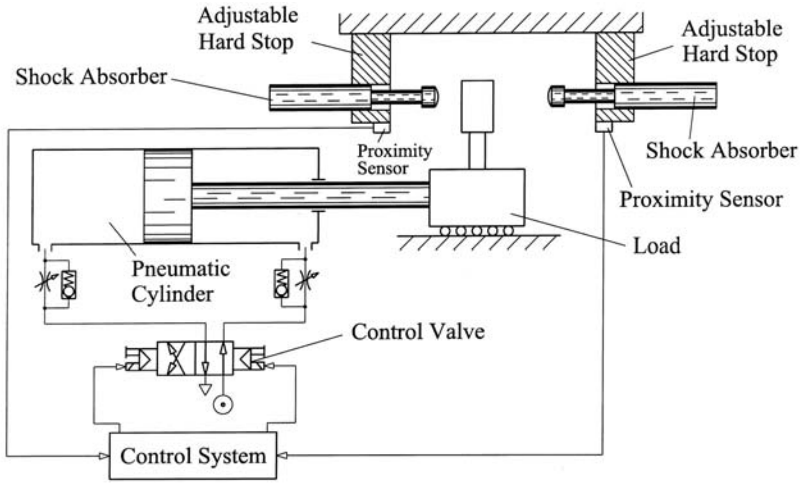


FIGURE 1.4
Schematic diagram of the pneumatic positioning actuator with two adjustable hard stops.

the magnetic field closes the switch, completing an electrical circuit and producing an electrical signal.

The basic function of the air cushioning is to absorb and dissipate the impact kinetic energy so that deceleration is reduced to a tolerable level. Linear and radial "float" of the cushion seals allows one to solve the problem associated with misalignment. Usually, adjustable air cushioning is used in the pneumatic cylinder if the piston velocity exceeds 0.2 m/s (second). Another major benefit of using air cushioning is that noise pollution, a hazard for workers and the environment, is greatly reduced. Because the contact surface at the stroke end is metal, stopping position repeatability is quite high (~0.01 mm).

Figure 1.4 shows the schematic diagram of the pneumatic actuator, which has the ability to stop the piston in the two adjustable positioning points within the whole piston stroke. The structure of this system is similar to the actuator illustrated in Figure 1.3, the main difference being the use of shock absorbers instead of the air cushions. This system can provide high speed (about 2 to 3 m/s) and positioning repeatability (up to 0.01 mm). The major weakness of this actuator is poor adaptability because the hard stops are not truly programmable. They must be moved manually to achieve a desired alternate position.

Shock absorber construction and parameters depend on the speed of the cylinder, the mass being moved, the external forces acting on the system, the system pressure, and piston diameter.

For implementation of the multiposition open-loop system, the so-called "multiposition pneumatic cylinder" is used, which typically consists of several connected cylinders (usually two or three). Figure 1.5 shows such an actuator with three pneumatic cylinders, which can reach four positioning

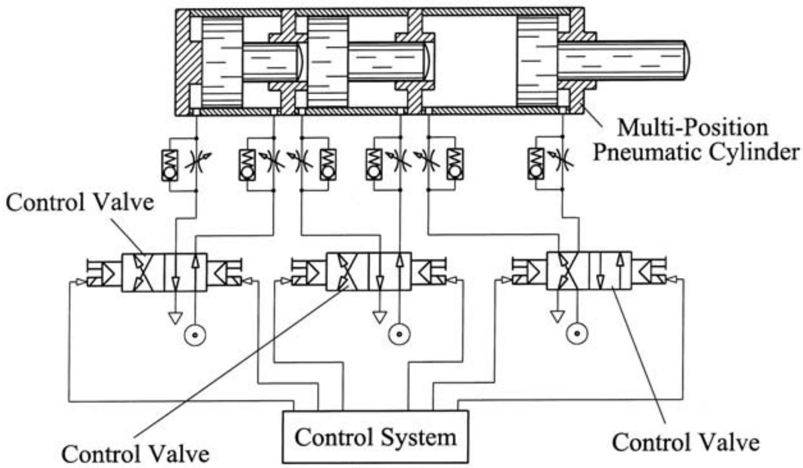


FIGURE 1.5
Schematic diagram of the actuator with multi-position cylinder.

points. The number of positioning points is defined by $N = n + 1$, where N is the number of positioning points and n is the number of connected pneumatic cylinders. Each rod that stands within the cylinder is the mechanical hard stop for the sequential piston, in essence; in this case, the two left pistons with their rods move the hard stop for the right piston. Although the construction is simple and affords high reliability, this actuator has impacts during the stop process that sometimes disturb the stability of the positioning. In addition, in a number of cases, such a system is bulky because of the numerous quantity of the control solenoid valves.

A similar positioning system with multiple stop points is shown in Figure 1.6. There is a positioning actuator with a so-called "digital" pneumatic cylinder, which consists of several pneumatic cylinders installed within the common sleeve. The stroke of the left-most cylinder is minimal, and each subsequent cylinder has double the stroke of the previous cylinder. Also, the rod of each cylinder is coupled to the body of the subsequent one that carries out the summation function for the cylinder's movement. Communication of the cylinder's pneumatic chamber with the supply pressure and exhaust line in variable combinations can achieve $N = 2^n$ positioning points with steps equal to the movement of the left cylinder. This allows using this construction not only as an actuator, but also as a digital converter. The use of diaphragm actuators provides more compact construction.

One can use the actuator depicted in Figure 1.7 for the implementation of multi-position open-loop systems with adjustable hard stops. Each sliding hard stop has its own actuator; in this case, there is a single-acting pneumatic cylinder, in which the rod is the hard stop. The sliding adjustable hard stops are assembled on a common base that can move along the main cylinder-moving axis. The base movement is limited by two mechanical hard stops

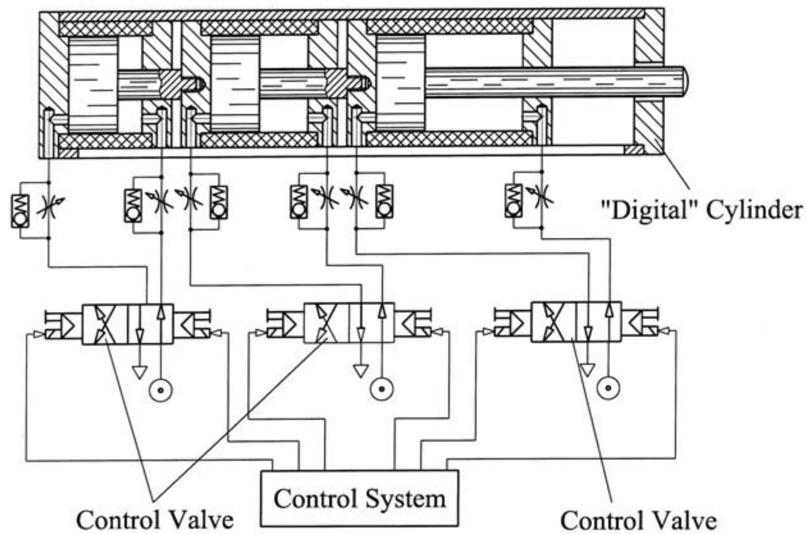


FIGURE 1.6
Positioning actuator with "digital" cylinder.

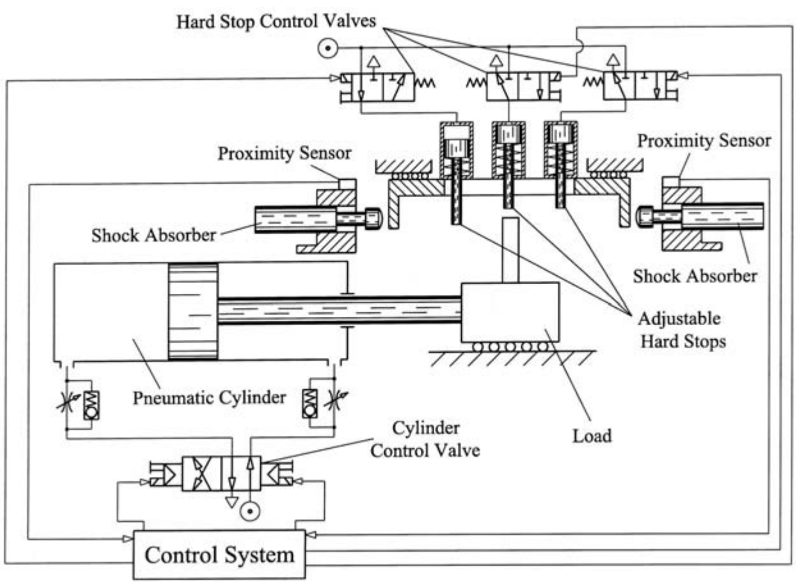


FIGURE 1.7
Multi-position actuator with hard stops.

with shock absorbers. Actuator stopping adjustment is achieved by mounting the pneumatic cylinders in the necessary positions. The pneumatic cylinder rods pass through the base slot. Usually, such a positioning actuator is used in cases where the number of the stop positions is not more than five or six; otherwise, actuator construction has a bulky build. The system positioning repeatability is approximately 0.03 to 0.04 mm.

The closed-loop pneumatic positioning actuator contains a transducer to measure and convert the actuator output signal to an electrical signal. This feedback signal is compared with the command signal, and the resulting error signal is applied to reach the necessary positioning or tracking of the movement. Two well-known technologies are widely used for point-to point closed-loop positioning systems: (1) airflow regulation using servo or proportional control valves and (2) a braking mechanism.

Usually, an actuator with a braking mechanism uses a pneumatically or electrically driven external mechanical brake, which consists of spring-loaded friction pads that act on the rod (or other moving component) of the pneumatic actuator. Typically, the application of air pressure causes the brake to release, providing hold actuation. Positioning is achieved in pneumatic braking systems by applying the pneumatic brakes at a predetermined point prior to reaching the target position. Braking is applied in an "on" or "off" manner, negating the possibility of programmable velocity control or a sophisticated deceleration profile.

The schematic diagram of such a positioning actuator is shown in Figure 1.8. It contains the pneumatic cylinder (1), a mechanical brake (2) drive by pneumatic cylinder (3), positioning sensor (4) that measures the load displacement, valves ($V_1 - V_5$), throttles ($R_1 - R_4$), and a control system. Four valves ($V_1 - V_4$) control the pneumatic cylinder (1), and they are arranged in pairs in series, which allows one to achieve independent adjustment of the high speed $\dot{x}_m$ and low speed (or creeping speed) $\dot{x}_c$ of the load. These

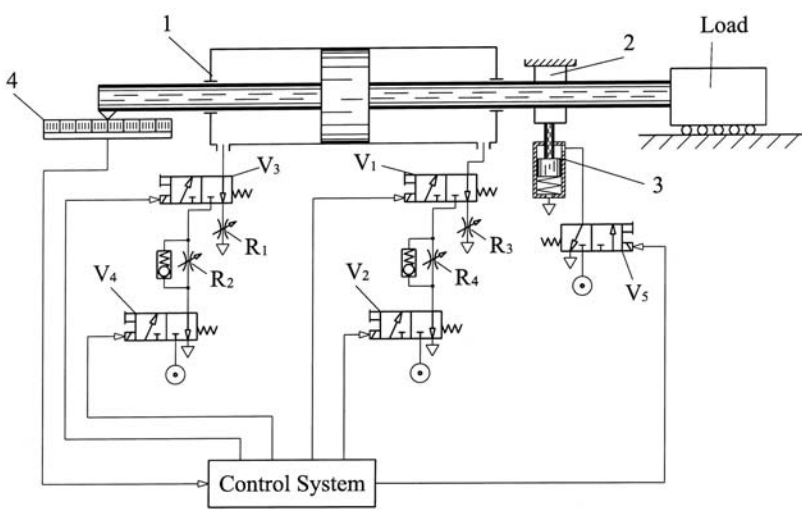


FIGURE 1.8
Positioning actuator with pneumatic brake.

adjustments are performed by four throttles (R_1 and R_3 for high speed, R_2 and R_4 for low speed). The valve (5) is used to control the brake pneumatic cylinder (3). To decrease the brake response time, a quick exhaust valve can be applied (not shown in Figure 1.8). In this case, the positioning (stop) process has two stages. In the first phase, the load speed is reduced from a high speed ($\dot{x}_m$) to a low (creeping) speed ($\dot{x}_c$) by pneumatic means. At the second stage, the mechanical brake is switched on and holds the load in the desired position. Figure 1.9 represents a typical velocity curve for this process and the control algorithm for this actuator, which is represented in Table 1.1. Here, x and $\dot{x}$ are the load position and velocity, respectively; V_i determines the valve's state (i = valve number: 1 = valve is energized, solenoid action; 0 = valve is deenergized, spring action); x_d is the coordinate of the positioning point (desired position); x_1 is the distance from the positioning point where the cylinder starts to change the velocity from $\dot{x}_m$ to $\dot{x}_c$; and x_2 is the distance from the positioning point where the brake is switched on.

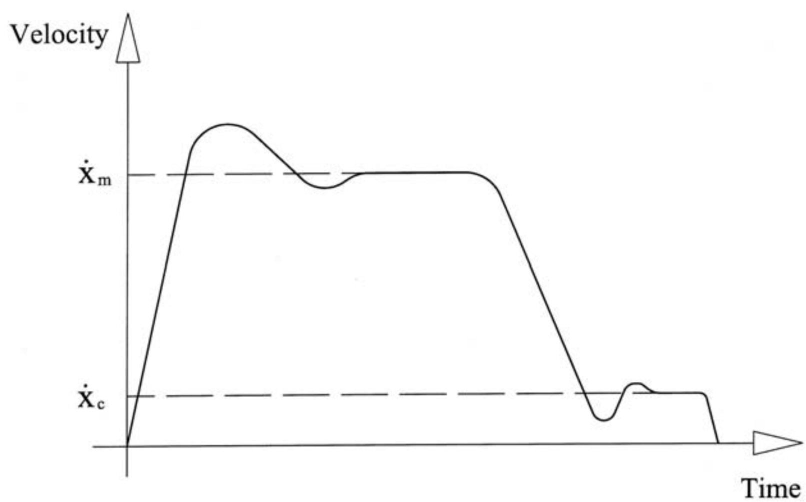


FIGURE 1.9
Velocity changing curve.

TABLE 1.1
Control Algorithm

x	V_i	$\dot{x}$	x	V_i	$\dot{x}$
$x \le x_d - x_1$	00111	$\dot{x}_m$	$x \ge x_d + x_1$	11001	$-\dot{x}_m$
$x_d - x_1 < x < x_d - x_2$	00111	$\dot{x}_c$	$x_d + x_2 < x < x_d + x_1$	11101	$-\dot{x}_c$
$x \le x_d - x_2$	11110	0	$x \le x_d + x_2$	11110	0

System tuning becomes sensitive to pressure variation, brake-pad wear, variation of the brake switching time, and the friction force in the pneumatic cylinder. Usually, the x_1 value is maintained constant, independent of any variation in the actuator operating conditions. The main criterion in its value selection is the nature of the transitional process — up to the moment of approaching the position $x = x_d - x_2$ or $x = x_d + x_2$, the oscillations of the creeping speed $\dot{x}_c$ must be negligible. Compensation for the influence of variation in operation conditions of the actuator is carried out by a change in the x_2 value. The simplest compensation algorithm has a correction for x_2, to be made in the next cycle, and is proportional to $(-\Delta_x)$, where Δ_x is the difference between the actual position of the stopping point in the preceding cycle and the preset position. Using this compensation algorithm, system positioning repeatability of ±0.15 mm can be achieved.[155,177]

Another positioning actuator involves placing magnetorheological braking devices functionally in parallel with a pneumatic cylinder or motor. Magnetorheological fluids are materials that respond to an applied field and the result is a dramatic change in rheological behavior. These fluids' essential characteristic is their ability to reversibly change from a free-flowing, linear, viscous liquid to a semisolid with controllable yield strength in milliseconds when exposed to a magnetic field. A typical magnetorheological fluid consists of 20 to 40%, by volume, relatively pure iron particles suspended in a carrier liquid such as mineral oil, synthetic oil, water, or glycol.

Magnetorheological brakes provide a braking force or torque that is proportional to the applied current. Through closed-loop feedback of the positioning sensor, accurate and robust motion control is achieved. The schematic diagram of such a positioning actuator is shown in Figure 1.10. The function of the three-position solenoid control valve is to ensure that the cylinder is

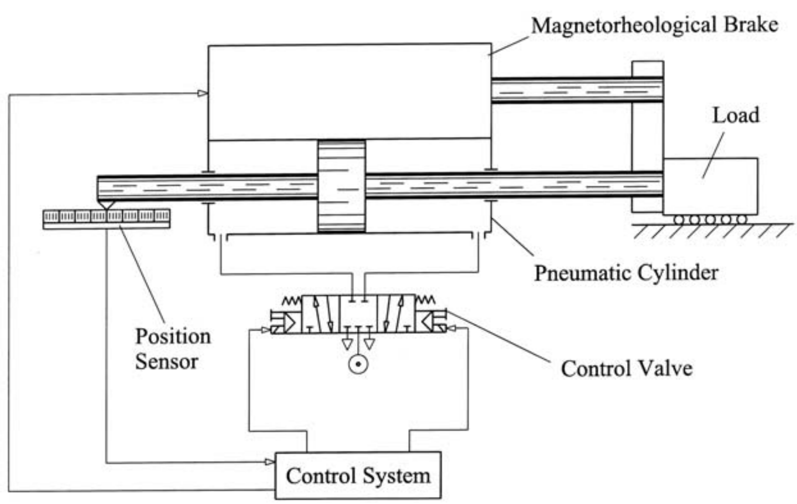


FIGURE 1.10
Positioning actuator with magnetorheological brake.

always directed toward the desired position and is commanded to a center position when the load is within some tolerance band around this point. Ideally, the magnetorheological brake function is substantial enough to "stall" the pneumatic cylinder. This will then provide the control authority with the ability to command a broad dynamic range of velocity control. Magnetorheological devices use a special liquid that undergoes property changes as a result of the action of the control current. In this sense, it is somewhat analogous to a conventional "air-over-oil" actuator. Because of significant hydraulic leveraging, this concept has a high force capacity.

The control algorithm can be of several types; one of them has the following description. Pneumatic logic is simple Boolean logic based on the sign of the position error. This logic also commands the valve to its neutral position when position x is within the tolerance band (Δ). The magnetorheological braking logic commands the application of braking (either step-wise or progressive) when position x is within Δ. The position is differentiated to provide an estimation of system velocity. System velocity is the basis for an error function that passes through a controller. This signal is summed with the magnetorheological braking signal to provide point-to-point velocity control.

This linear positioning system has a wide range of velocity control; for example, an actuator with pneumatic cylinder of 32-mm diameter bore and 160-mm stroke has the ability to move with constant velocity from 20 to 500 mm/s; the system positioning repeatability is ±0.15 mm.[89]

A positioning actuator is usually used for the servo or proportional control valve, which achieves the desired position by regulating the volume and flow rate of air into and out of pneumatic actuators. This linear positioning system (Figure 1.11) comprises the pneumatic cylinder, the positioning sensor (transducer), an electronic control system, and a continuously-acting valve as the control element.

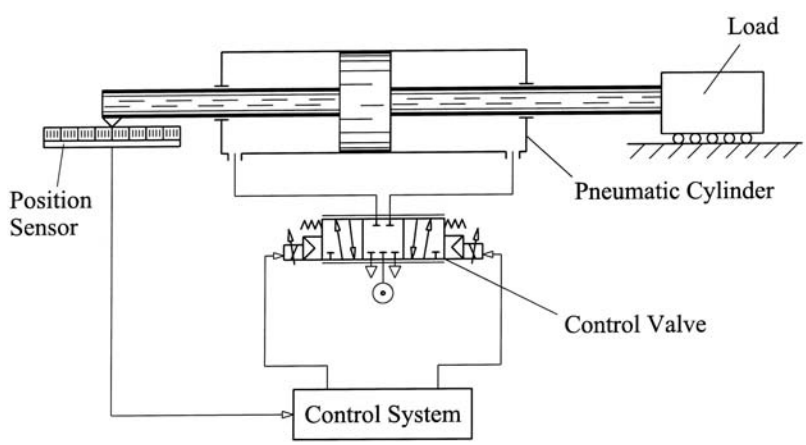


FIGURE 1.11
Positioning actuator with servo or proportional control valve.

Accurate, high-speed actuator movement requires quick and exacting valve response to commands from the control system. As the pneumatic cylinder approaches a set point, the valve shifts over-center to build up pressure that opposes piston motion. Internal algorithms control rapid shifting of the valve from one side to the other, giving smooth deceleration to the required position with the necessary dynamic characteristics.

Control algorithms provide the key to making positioning servo pneumatics work. As in most closed-loop systems, velocity and acceleration are controlled using three-loop position feedback (state controller or PVA controller). Three-loop algorithms measure position directly from the feedback transducer. Velocity and acceleration are derived from the position vector. The controller sums up these three different signals and generates a correction signal to the valve.

A practical consideration is that any servo system must be tuned. In this case, the control system calculates baseline loop parameters for stable operation based largely on the type of control valve and cylinder, as well as the payload, and motion parameters. The adaptive control algorithms also measure the quality of motion after every cycle to constantly optimize performance. For example, if overshoot is too high or low, it adjusts the filter parameters to improve response. Self-tuning also comes into play when the payload suddenly changes or seal and bearing characteristics change with use. Gain adjustments, critical damping, and overall system sensitivity can also be set manually.

A positioning actuator with a servo or proportional control valve can operate both the point-to-point modes and tracking motion.

In point-to-point mode, a velocity profile usually has a "trapezoid" form. In this case, the acceleration and motion with high constant velocity is realized by switching off the control valve to the saturation position. Only around the positioning point does the control valve move within the regulation range that provides the deceleration and stop process in the desired position.

In the tracking motion mode, the control valve permanently operates in regulation range, and valve effective areas change until the load stops.

For both tracking and point-to-point positioning, high performance control has nearly the same meaning: fast and accurate response to the reference. However, for tracking, the concern focuses on the response behavior along the entire reference trajectory, while for point-to-point positioning, the concern focuses on the response behavior around the reference point. Basically, high-quality, point-to-point implies high-quality tracking. It is well known that in the presence of uncertainly and disturbance; the point-to-point positioning quality is primarily decided by the feedback control quality. Well-designed feedback control will directly give high-performance, point-to-point positioning and facilitate the tracking control performance improvement.

In these working modes, the friction force in the pneumatic actuator plays a very important role. Friction will cause a steady-state error in point-to-point positioning and a tracking lag in a tracking motion. For precision tracking motion and positioning, the significant adverse friction effect must be compensated for.

Another important factor is the control valve nonlinearities (hysteresis, valve friction, dead zone, variation of the flow coefficient) that decrease the actuator positioning accuracy and dynamic characteristics. In part, this problem can be solved using a fast solenoid control valve (instead of the servo or proportional valve), which operates with PWM or Bang-Bang controller. In this case, if chattering exists, it may act as a dither, which is a classic friction compensation technique.

1.2 Pneumatic Systems for Velocity Control

The application fields of servo pneumatic actuators with speed motion control include arc welding machines; painting and printing equipment; scanning motion systems in inspection devices; cutting machines for plastic, wood, and fabric materials; gluing; and others.

In practice, open-loop pneumatic actuators are seldom used in these applications because of the poor ability to maintain constant velocity stabilization owing to low internal damping, high sensitivity to load and friction force changes, as well as the actuator's nonlinear characteristics.

The pneumatic actuator with magnetorheological braking devices (Figure 1.10) can also be used in velocity control systems. For example, the linear system with a pneumatic cylinder of 32-mm diameter bore and 160-mm stroke has the ability to move with constant velocity from 20 to 500 mm/s. In this case, the control accuracy is about 10% of the programmed value.

Figure 1.12 is a schematic diagram of the rotary pneumatic actuator, which has the ability to control the rotation velocity. The function of the three-position solenoid control valve is to ensure that the motor is rotated in the desired direction (clockwise or counterclockwise). The motor is stopped when the valve is at the center position. The actuator rotates the shaft, on which the magnetorheological brake, load, and velocity sensor are installed. Changing the brake impedance torque controls the shaft's angular velocity. For this system, the control accuracy is about 15% of the programmed value.

A pneumatic actuator for velocity control with a servo or proportional valve is based on the same principle of error-signal generation as the positioning servo actuator, except that the velocity of the output is sensed rather than the position of the load. When the velocity loop is at correspondence, an error signal is still present and the load moves at the desired velocity.

Most pneumatic servo applications require position control in addition to velocity control. The most common way to provide position control is to add a position loop "outside" the velocity loop, which is known as cascading

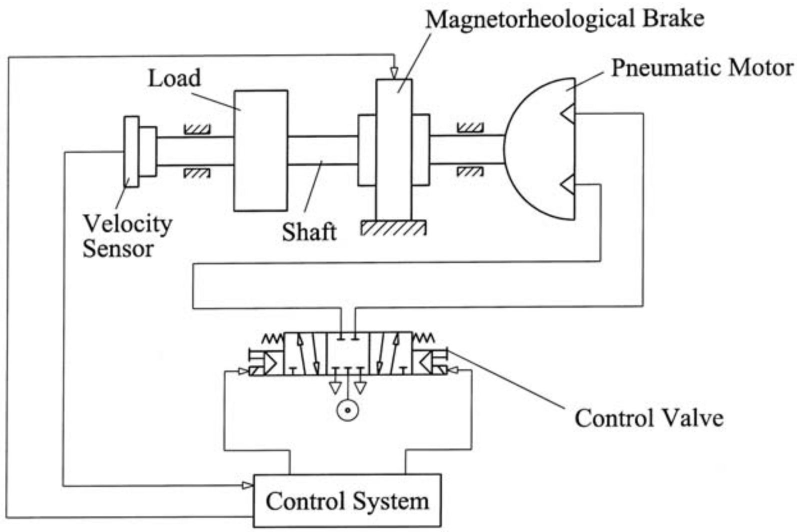


FIGURE 1.12
Rotary actuator with magnetorheological brake.

loops. In this case, the position error is scaled by the position loop gain to produce the velocity command.

Ripple, linearity, and low-speed performance are generally the most important characteristics of the velocity transducers or an algorithm that allows reaching the velocity signal because these characteristics determine the static errors in the velocity servo.

Figure 1.13 shows the schematic diagram of the linear pneumatic actuator with a solenoid valve for velocity control. Basic components include a standard rodless cylinder (1) with adjustable end-position cushioning at both ends; and the control valve (2), which is a standard double-solenoid valve with closed-neutral position and two flow positions (5/3-way valve). The position transducer (3) measures the load displacement, and the control system determines the velocity command and forms the control signal for the valve (2) according to the control algorithm. Using a nonreturn valve (4) in the supply port allows for energy recuperation in the motion reverse process; in this case, the kinetic energy of the moving mass is used for its acceleration in the opposite direction. Using two piloted nonreturn valves (5) with two single solenoid valves (6) results in an increase in the actuator efficiency because in this case the additional exhaust lines allow for an optimal ratio between the effective areas of the supply and exhaust port, which improves the steady-state velocity accuracy and the acceleration (deceleration) process.[102] For effective performance, the nonreturn valves (5) with solenoid valves (6) must be fitted directly into the ports of the pneumatic cylinder (1).

This system operates with PWM (pulse width modulation) or Bang-Bang controller with a four-loop feedback algorithm, which measures the load position directly from the feedback transducer; and velocity, acceleration,