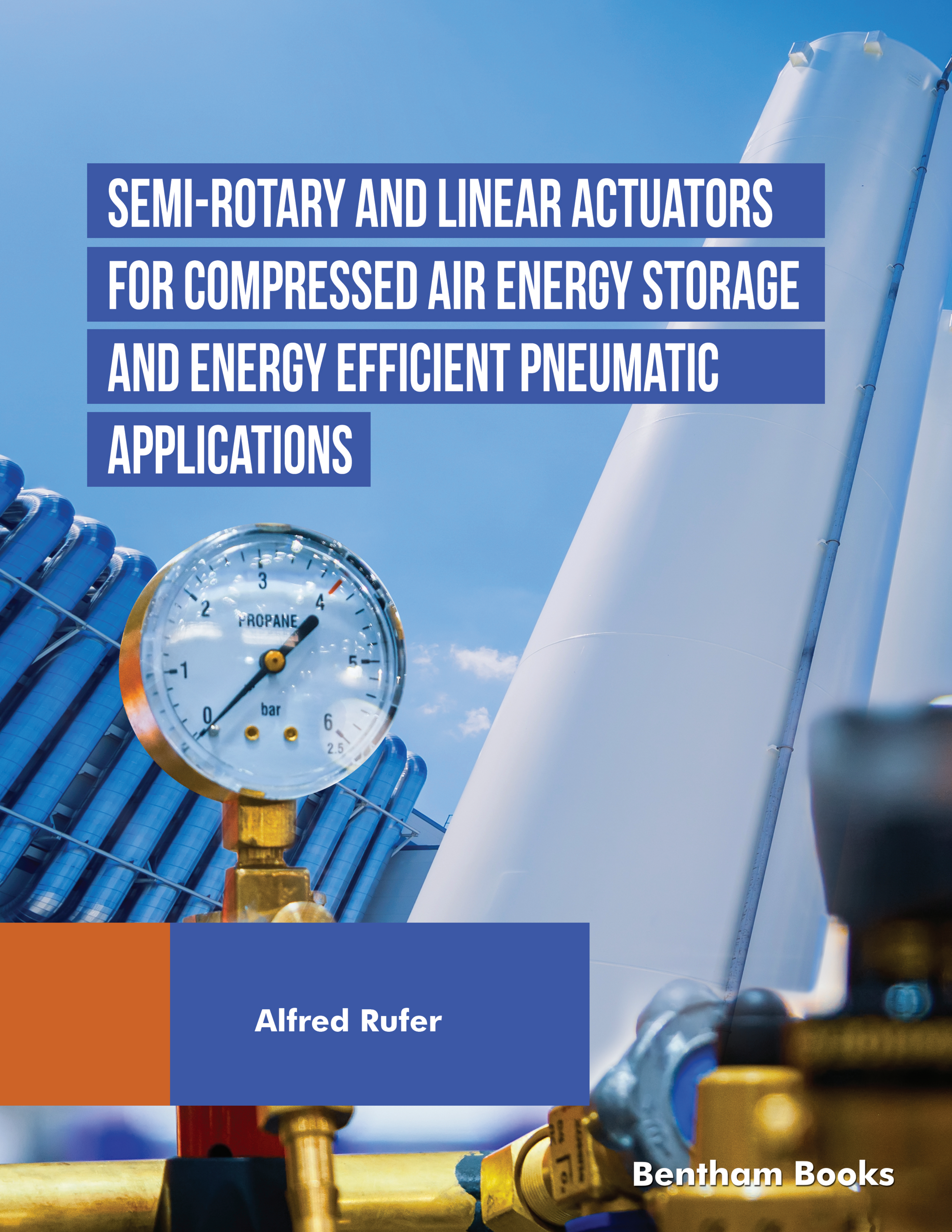
SEMI-ROTARY AND LINEAR ACTUATORS FOR COMPRESSED AIR ENERGY STORAGE AND ENERGY EFFICIENT PNEUMATIC APPLICATIONS



Alfred Rufer

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Semi-rotary and Linear Actuators for Compressed Air Energy Storage and Energy Efficient Pneumatic Applications

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PREFACE

In the context of the many challenges to society related to energy and environmental issues, the utilisation and the storage of electrical energy appear at a front level of needed industrial developments, accompanied by academic research and other investigations.

The technology of compressed air is a simple and reliable technique widely used in the sector of industrial handling and actuators but has recently become an attractive means for energy storage in different forms. The main argument behind the use of compressed air energy storage is given by the use of simple mechanisms issued from reversible physics in comparison to electrochemical principles, where the calendric and cycle ageing mechanisms have been the centre of questions for many years. The question of recycling elementary materials is another aspect related to the battery industry and public services.

Regarding the sustainability aspects of the use of energy, the general question of efficiency is now the centre of many considerations worldwide, and more and more studies and comparisons are made at the system level, where the different individual or cascaded energetic transformations are evaluated. A strong example comes from the automotive sector, where the classic ICE (Internal Combustion Engine) vehicles with their reservoirs are compared to Hydrogen powered vehicles with fuel-cells, or further with BEV (Battery Electric Vehicles).

Back to the technique of compressed air, the industrial world uses from long time pneumatic actuators for their simplicity, reliability and low costs. But regarding the energetic balance, this technology presents, in its actual form, many disadvantages that can be qualified as energetic aberrations. And the use of pneumatic devices for the transformation from compressed air energy to mechanical and electrical power must be reconsidered.

This book tries to give answers to the questions of the energetic efficiency of pneumatic devices and tries to use new arrangements for an application to energy storage. When speaking about energy storage, the question of the reversibility of the transformations or energy flows is also addressed. Even when the actual or classical industrial pneumatic devices are not foreseen for an operation as compression stages, the principle of using them as such is considered, and will need adaptations of those devices, especially at the level of their sealing elements.

The compressed air energy storage principle is used in the industrial world in the form of air reservoirs used as buffers feeding the pneumatic actuators and motors. Here the buffering function serves to power devices with a strong flow of pneumatic energy, and normally, pressure regulating valves are rarely used. But several proposals are made in the sense of using compressed air stored at a higher level of pressure and with an adaptation element to the application. The properties of such pressure reduction elements are also discussed in this book.

Further in the direction of realizing compressed air energy storage, a low pressure storage system called the underwater compressed air energy storage (UWCAES) is described and represents one of the ways for storing energy and using pneumatic converting elements for which the actually used pressure level fits the UWCAES system.

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CHAPTER 1

Introduction and Summary

Abstract: The motivation for the use of compressed air as an energy carrier and as a storage means for many industrial applications resides in the simplicity and low-cost conditions of its implementation. However, conventional pneumatic technology suffers from a very low energetic efficiency even if the production, use and recycling of the components can be said to be environmentally friendly, and it does not use problematic materials. This introductory chapter positions pneumatic technology and discusses a possible extension of the applications to the sector of energy storage in a general manner. The chapter gives the historical background of the presented developments of the book and gives an overview of the content of the document.

Keywords: Energy storage, Efficiency, Low pressure storage, Pneumatic actuators, Pumped hydro, Battery energy storage, Ageing effects.

1. INTRODUCTION

During the second half of the 20^{th} century, many questions arose about the availability of fossil energy resources and about the greenhouse gas emissions linked to their combustion. These questions have prompted several efforts in the development of alternative energy sources, mainly photovoltaic systems and wind energy.

The intermittent nature of these sources linked to day-night alternation and seasonal or weather conditions has triggered new developments downstream in the sector of energy storage technologies.

As a complement to well-established storage facilities like pump-turbine hydropower plants, battery energy storage systems can be considered very well suited for supporting decentralized power producers.

Successive iterations of battery technology have shifted battery applications from the older lead-acid or nickel-cadmium or nickel-metal-hydride cells to the new lithium-ion technique, which features higher energy and power densities and allows the realization of applications under much better economic conditions.

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Generally, for electrochemical batteries, the question arises of the materials available in the future if their development evolves in the direction of very large volumes. The particularly concerned areas of future electrical systems are distributed generation and electrical mobility.

Not only the material resources available but also the indirectly related topics of the global life cycle and aging phenomena are becoming increasingly important, as well as the still open questions on the recycling of all components and materials used in the manufacture of an electrochemical accumulator.

In the context of sustainable energy strategies, several alternative solutions for energy storage are investigated as technologies based on reversible physics like mechanical or thermodynamic principles.

Compressed air energy storage (CAES) can be considered a potential solution, using only standard materials and established technology. Additionally, and in opposition to the electrochemical batteries, these systems can be repaired or refurnished, offering unbeatable longer life cycles. Another advantage of CAES is that their materials are not problematic for recycling [1 - 4].

The development of CAES systems includes the development of high-performance compression and expansion machines and must comply with the elementary rules of thermodynamics.

By many different development projects, the focus has been set on isothermal compression and expansion [5 - 7], with the goal to reach the highest possible efficiency. Also elementary conversion means based on classical pneumatic equipment have been proposed, where the operating principle has led to limited performance.

If the classical pneumatic devices are generally classified in the category of low efficiency devices, they present the advantages of limited costs. Regarding their efficiency, several solutions have been proposed, such as adding an expansion chamber to the original displacement volume in order to recover a significant part of the fluid's enthalpy [8 - 10].

For the same category of classical pneumatic devices, the normal operating pressure is in the order of tens of bars. Using them in the context of CAES will have the consequence of strongly limiting the system's energy density if the storage reservoir is designed for the same pressure level as the pneumatic converters. However, one possibility exists where the storage pressure is of limited value. This is the so-called Under Water CAES, where the reservoir consists of immerged bags with a highly deformable volume. Such systems can be placed underwater at an immersed depth of one or two hundred meters, leading to a storage pressure compatible with the low pressure of classical conversion devices [11 - 12].

Another advantage of UWCAES is that they can be operated under constant pressure for the whole range of their storage capacity.

Last-but-not-least, the specially designed energy bags for UWCAES have the property of needing only very low energy for their realization, leading to storage equipment with very low grey energy [13].

In this book, proposals are made for the enhancement of the energy efficiency of systems based on classical industrial pneumatic devices used in energy storage based on low pressure. The main contributions concern the expansion process, where linear and rotational actuators are used as prime movers of an electric generator. Then, with the aim to use the same components in the compression process, the reversibility of the components and systems is analysed and measured.

Regarding the use of vane-type rotational actuators, the enhancement of efficiency is proposed for an original system called the Gallino system, where an oscillating angular actuator drives the generator with the help of a so-called motion rectifier [14 - 15]. The main contribution concerns the pneumatic to mechanic conversion, where in addition to the classical displacement work of the actuators, an expansion volume is added to the system allowing to recover an important part of the primarily injected enthalpy.

Additionally, the influence of a pressure regulation valve on global efficiency is discussed. Such a valve is used in the Gallino system as a pressure reduction element between the storage reservoir and the pneumatic actuator.

Because the motion rectifier in the Gallino system does not allow the reversibility of the power flow, a solution for the interface between angular actuators and the electric generator is proposed. This solution is based on a crankshaft and connecting rod assembly. A two-channel system with two 90° shifted actuators is proposed, allowing low speed operation and starting from any angular position.

Then, the study describes a generator drive using classical linear pneumatic cylinders. First, the operation and energetic performance of a single cylinder is simulated and calculated. In this system, the back-and-forth movement of the piston is transmitted to the generator through a classical crankshaft.

CHAPTER 2

Compressed Air Systems and Storage

Abstract: The elementary principles related to compressed air are presented, describing the basic compression and expansion characteristics. The adiabatic, polytropic and isothermal phenomena are described together with the definitions of the energy content of a given volume. Different loss factors related to compressed air are enumerated together with the advantages and drawbacks of pneumatic technology.

Then, the possibility of storing energy under low pressure conditions as the so-called Underwater CAES system is discussed. Such systems have the interesting property of being realized with a very low amount of grey energy.

Keywords: Adiabatic characteristic, Compression of air, Expansion of air, Energy content, Isothermal characteristic, Loss factors, Low pressure storage, Properties of energy storage, Polytropic characteristic, Pressure reduction valve, Underwater CAES.

1. THE PHYSICAL PRINCIPLES RELATED TO COMPRESSED AIR

A compressed air energy storage system is based on elementary principles of thermodynamics [9]. According to the general scheme of compressed air energy storage, two main types of components are used, namely compression/expansion machines, where mechanical work is the main input vector, as well as storage vessels, where the mechanical work is equal to zero. The two types of components are represented in Fig. (**2.1**).

Both components can be considered as separate control volumes and respond to the rate form of the First Law of thermodynamics.

$$\dot{W}_i = \dot{Q}_e + \dot{m}_e \cdot h_e - \dot{m}_i \cdot h_i + \frac{dU}{dt} \tag{2.1}$$

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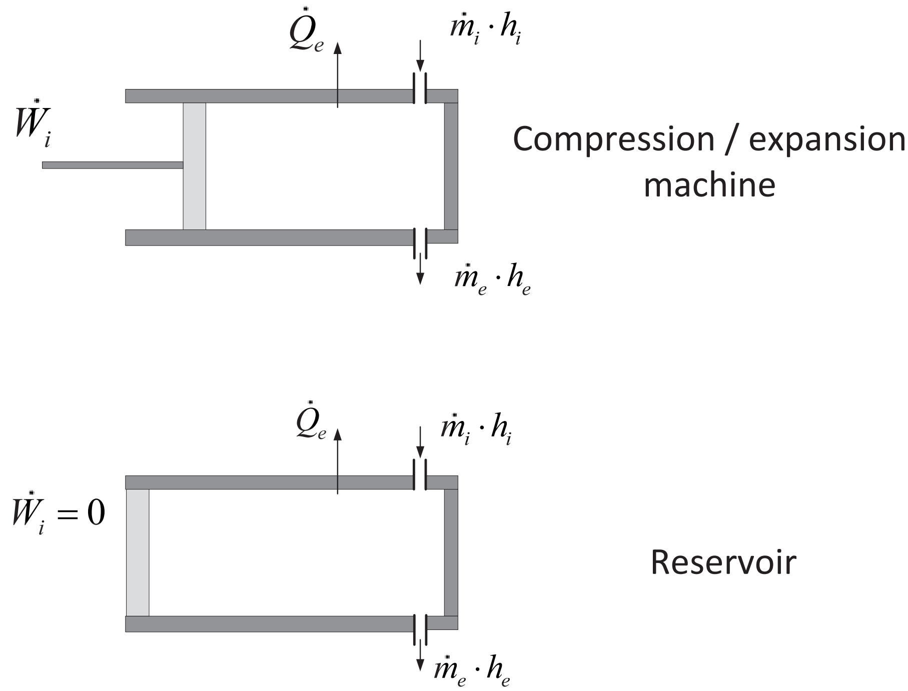

Fig. (2.1). Main components of a CAES.

$\dot{W}_i$ and $\dot{Q}_e$ are the work and heat flows transferred to the gas from the external environment, $\dot{m}_i$ and $\dot{m}_e$ are the input and exit mass flows, and h_e , h_i are the corresponding specific enthalpies [J/kg].

The compression and expansion machine's main function is to change the thermodynamic state of the gas inside the control volume, and further to maintain the input and exit flow rates.

The reservoir itself is characterized by the absence of work transferred to the gas (= 0).

The description of a compression machine is based on the assumption of the air being considered as an ideal gas (PV = mRT), and further on the basic relations for the work and the heat.

$$W = \int p dV \tag{2.2}$$

$$\dot{Q} = H_c A_c (T - T_w) \tag{2.3}$$

H_c is the heat transfer coefficient, A_c is the cylinder surface area exposed to convection, T_w is the temperature of the surface area and T the instantaneous gas temperature.

For a compressor, assuming steady state conditions where no energy is accumulated in the device, the following relation can be written as a combination of the First and Second Laws [16].

$$\dot{W} = (1 - \frac{T_0}{T})\dot{Q}^{-} + \dot{m} \cdot \psi \tag{2.4}$$

Where ψ is the flow energy defined by:

$$\psi = (h - h_0) - T_0 (s - s_0) \tag{2.5}$$

h and *s* are the specific enthalpy and entropy, and the subscript 0 indicates that the properties are taken at reference temperature and pressure (T_0 =20°C, p_o = 1bar).

The exergy flow (usable energy) of the produced air stream is then expressed as:

$$\dot{X} = \dot{m}[h - h_0 - T_0 (s - s_0)] \tag{2.6}$$

In the case of an ideal gas flow:

$$h - h_0 = c_p (T - T_0) \tag{2.7}$$

$$s - s_0 = c_p \ln \frac{T}{T_0} - R \ln \frac{p}{p_0} \tag{2.8}$$

The air stream exergy can be split into two parts, the pneumatic part and the thermal part, as follows:

$$\dot{X} = \dot{X}_{(pn)} + \dot{X}_{(th)} \tag{2.9}$$

CHAPTER 3

Increasing the Energetic Efficiency of Pneumatic Devices

Abstract: The chapter presents the main principle on which the proposals of this book are based. In this principle, the energetic efficiency of pneumatic actuators is strongly increased by adding an amount of expansion work to the classical work produced by constant pressure displacement. Such a principle has already been applied in steam machines at the beginning of the 20th century or in existing pneumatic converters used as motors for automotive vehicles.

Keywords: Constant Pressure Displacement Work, Expansion Work, The Truglia Motor, Compressed Air Car.

1. RECOVERY OF THE PNEUMATIC ENERGY

In Section 2.3.2, the pressure-volume diagram related to the recovery of pneumatic energy is shown in Fig. (**3.1**) and the elementary principle is described.

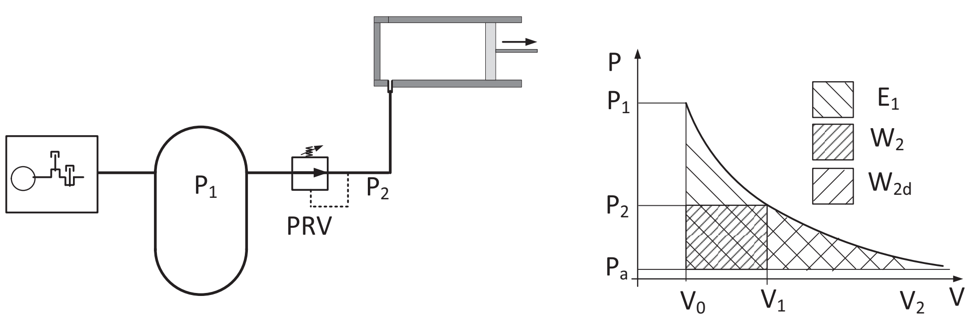

Fig. (3.1). Energy recovery of the pressurized air.

In the case of the operation of a classical pneumatic cylinder, the mechanical work is produced only by the displacement of the piston under the condition of a constant pressure P_2. At the end of the stroke, the pressurized air is released to the

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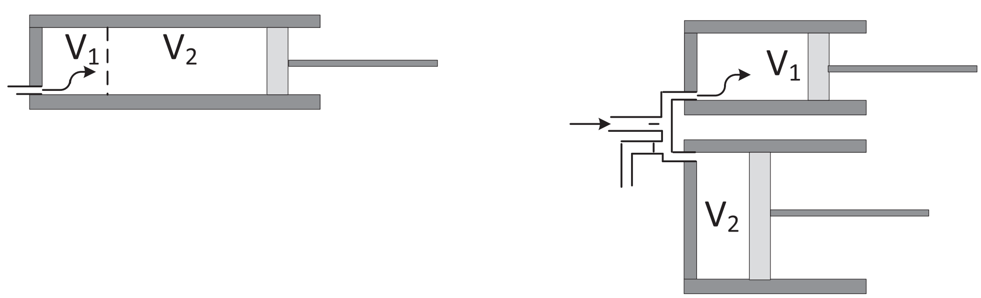
external by opening an exhaust valve. In order to increase the resultant efficiency, the pressurized air of the cylinder's chamber should be expanded within an additional step of the process.

For such an expansion, two categories of principles will be described. For the first category, the constant pressure displacement and the expansion of the air occur in the same volume or in the same cylinder. Fig. (**3.2a**) illustrates this principle where the V_1 volume corresponds to the volume with constant pressure and the sum of V_1 and V_2 to the volume of the expanded air. The change between the constant pressure work and the expansion is controlled by closing the intake valve when a volume V_1 of air has been intaken. The principle of expanding the fluid had already been applied in steam machines at the beginning of the 20th century with the goal of reducing the consumption of steam [19]. A more recent example of such a process is given by the principle of the so-called Truglia motor [20]. The operation principle of the Truglia motor will be described in Section 1.1.



a) b)

Fig. (3.2). Adding an expansion volume **a**) in the same cylinder, **b**) in an additional cylinder.

In the second category, the constant pressure displacement is realized in a first cylinder, and the expansion of the air is realized through a transfer of the air into a second cylinder Fig. (**3.2b**). This principle has been used for the MDI motor of the compressed air car [21]. This principle will be described in section 3.1.2.

In this second category, the principles of sequential and reciprocating strokes will be discussed. The MDI motor is a sequential strokes machine (Section 3.1.2), the other principle of the reciprocating strokes is presented in a short way in Section 3.1.3. Further, this principle will be analyzed in more detail through two applications in Chapter 4 and 5. The first of these applications corresponds to improving the efficiency of vane-type rotary actuators. The second application corresponds to a pneumatic motor realized with two double effect linear actuators.

1.1. Operating Principle, Defaults and Improvements of the Truglia Motor

The pneumatic motor invented by Vito Truglia uses a classical base of an ICE motor, pistons, and crankshaft, together with all auxiliaries and a mechanical transmission system. Only the upper part, namely the cylinder-head is modified with its distribution components.

The three main components of the Truglia motor cylinder-head are first an inlet valve controlled by the piston itself in the position around the upper dead center. Second, an exhaust valve controlled by a specific cam tree. The opening of the exhaust valve is done between the lower dead center and the opening of the inlet valve, allowing the exhaust of the expanded air. Finally, an anti-return valve is integrated into the cylinder-head in order to avoid the production of a negative torque when the expansion of the air reaches a level under the atmospheric pressure due to low inlet pressure. Fig. (**3.3**) shows the three valves of the Truglia motor.

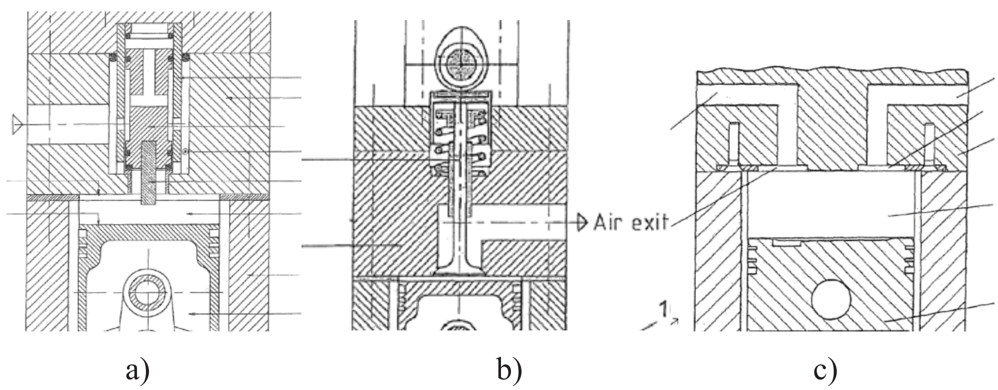


Fig. (3.3). The three specific valves of the Truglia motor **a**) Inlet valve, **b**) Exhaust valve, **c**) Anti-return valve

The position of the piston and the openings and closings of the valves are indicated in Fig. (**3.4**). In this figure, the horizontal axe corresponds to the time, under the condition of a mechanical frequency of 31.4 rad/s. The period of the piston cycle is equal to 200 ms.

CHAPTER 4

Coupling Two Rotary-Type Actuators

Abstract: This chapter presents the first example of the combination of two actuators of different volumes with which the principle of adding an expansion work can be realized. The two semi-rotary actuators are mechanically coupled and describe an oscillatory motion. Then the oscillatory motion is transmitted to an electric generator through a so-called motion rectifier. The structure of the new system is presented with the control valves and control circuitry. The different variables of the system as pressure, torques, and mechanical work are calculated by simulation. The efficiency of the new system is calculated and compared with the efficiency of a single actuator without expansion. The principle of adding an expansion work with semi-rotary actuators is then presented but with one actuator only where the expansion occurs in the same and unique chamber. Efficiency, torque waveform and produced mechanical work are presented, as well the control circuits.

The power reversibility of a system using semi-rotary actuators is addressed, and a solution with a crankshaft is studied.

Keywords: Adiabatic and Isothermal Expansion, Expansion Work, Power Reversibility, Semi-rotary Actuators, Torques.

1. CONTEXT AND MOTIVATION

In the introduction (Chapter 1), the System Gallino was introduced. This air-powered diving lamp uses one rotary actuator for pneumatic to mechanical conversion. As was described, a so-called motion rectifier transforms the alternating movement of the actuator into a unidirectional rotative motion, driving an electric generator. The general principle is maintained in this chapter, but in order to improve the energetic efficiency, a second actuator of the same type but with a larger volumetry is directly coupled to the original one. The chambers of the vane-type machines are moving in a synchronous tandem operation.

1.1. Structure of the System

Fig. (**4.1**) shows the global scheme of the proposed new system. As pneumatic to mechanical converters, vane type rotary actuators are used [21]. Such actuators

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have an alternating rotary movement of 270°. In this concept, the two actuators are directly coupled, and their alternating movement is transformed into a fully rotating one using a mechanical motion rectifier, as will be described more in details in Section 4.1.3 After rectification, the output shaft is coupled to an electrical generator.

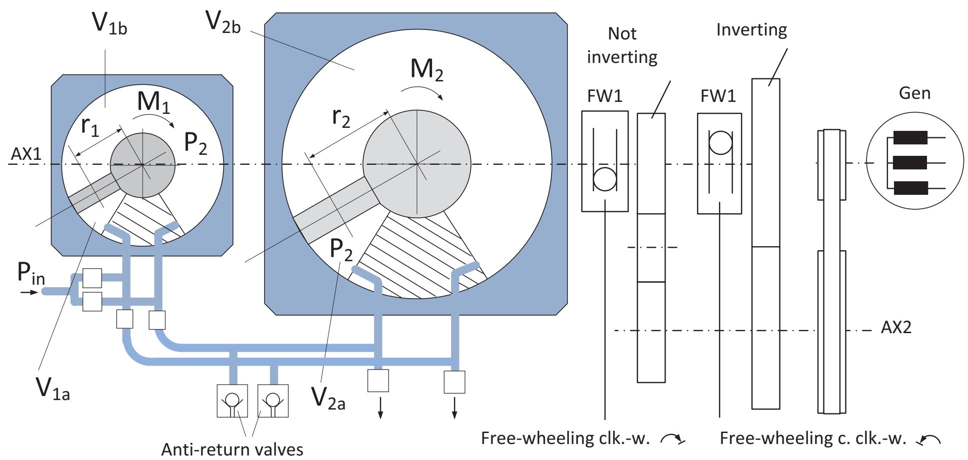

Fig. (4.1). The concept of the compressed air driven generator.

The two vane-type rotary actuators have two active chambers each, corresponding to the volumes V_{1a} and V_{1b}, respectively V_{2a} and V_{2b}. The chambers V_{1a} and V_{1b} are fed alternatively by the input compressed air, and they produce torque contributions alternatively according to the two clockwise and anti-clockwise motions.

1.2. The Mechanical Motion Rectifier

The wing-rotors of both actuators are mounted synchronously on the same shaft AX1 Fig. (**4.1**). This shaft transmits its alternating motion *via* two one-way (anti-return) roller clutches coupled to an inverting and a non-inverting gear to an output shaft AX2, resulting into a unidirectional full rotative motion [15]. From this output shaft, the motion goes to the electric generator *via* another multiplying gear. This additional gear is foreseen for an adaptation of the slow motion of the actuators to a sufficiently high rotational speed of the generator.

1.3. Operating Principle

The chambers V_{2a} and V_{2b} are fed from the exhaust air of the chambers of the first actuator. Volume V_{2a} receives the exhaust air of the V_{1b} chamber during the clockwise motion, and the volume V_{2b} that-one of the V_{1a} chamber during the anti-clockwise one. Because of the different volumes of the chambers of the first and second actuators, the air-transfer from the chambers of the first actuator to that-ones of the second-one corresponds to a real expansion of the transferred air, allowing so to recover a significant part of the internal energy of the compressed air. In the studied example, the volume ratio of the two actuators is chosen as $V_2/V_1 = 3$.

The schematics of the system with its control circuits are represented in Fig. (**4.2**).

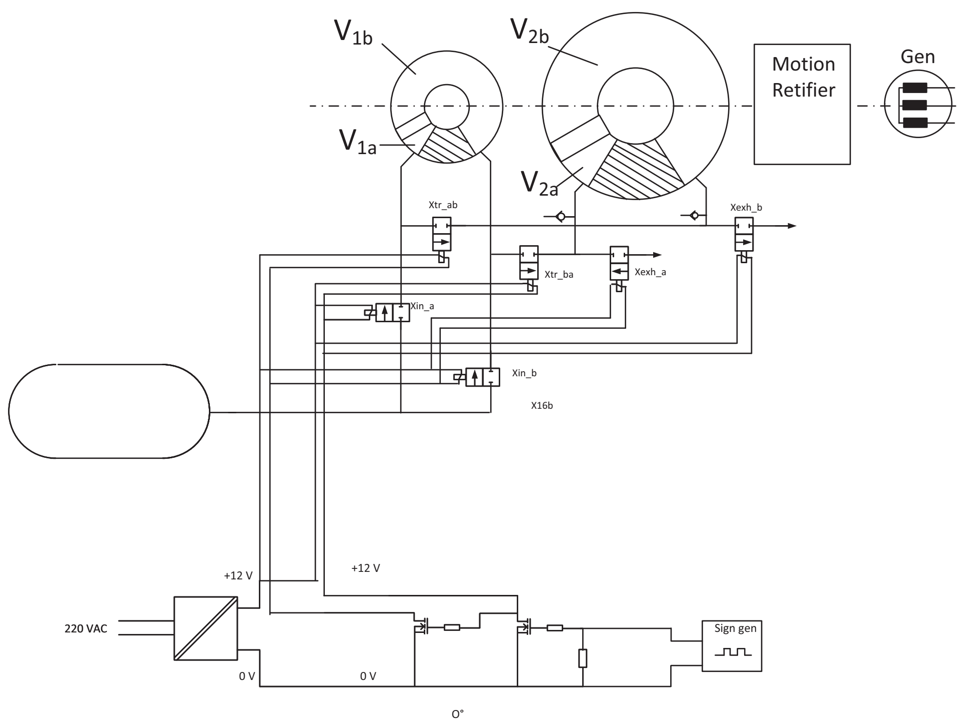

Fig. (4.2). Control valves and control circuits of the system.

For the control of the different airflows, 6 valves are needed. First, the air is controlled between the supply reservoir and the chambers of the small actuator

CHAPTER 5

The Pneumatic Motor with Linear Cylinders

Abstract: A pneumatic motor is studied where the pneumatic actuators consist of linear cylinders. This mechanical principle based on the use of a crankshaft and piston rods has the inherent property of being reversible. For this system, the same principle of adding expansion work with an additional volume is applied as it was in the previous chapter for semi-rotary actuators.

The mechanical behavior of the crankshaft and piston rod is described, and the following pneumatic displacement and expansion work is simulated. Two different architectures are simulated, namely, first, a system with pistons operating in phase and second, with alternating pistons. The energy efficiency of the new motor is calculated and compared with the efficiency of a system using a single linear cylinder.

Further, the principle of realizing the expansion work in the same cylinder as for the displacement work is applied to the motor with linear cylinders. The torque, power and converted work are presented with the simulation results. The study is completed with the presentation of a physical demonstrator system.

Keywords: Adiabatic expansion, Alternating piston-rods, Energy efficiency, Expansion work, In-phase piston rods, Linear cylinders, Pneumatic motor, Torques.

1. BASIC PRINCIPLE

A pneumatic motor can be realized on the basis of a linear piston/cylinder component. In such a system, the linear movement of the piston is converted into rotational motion using a conventional crankshaft and connecting rod.

Based on the previous descriptions of principles used to increase energetic efficiency, a pneumatic motor concept using two linear cylinders is described. In this concept, the linear cylinder pistons are connected to a classical crankshaft with crankshaft pins shifted at 180°. This concept belongs to the previously defined category of motors with expansion in an additional chamber and reciprocating strokes (Section 3.1.3).

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In section 5.2, the dynamics of the piston/crankshaft assembly are given, with relations to calculate the torque and the lateral reaction force.

In section 5.3 of this chapter, a simple pneumatic motor is described where the cylinder works according to the classical principles of pneumatic devices, namely with only a succession of sequences with constant pressure displacement work. The typical variables are simulated as the dynamics of the piston/crankshaft assembly and the developed torque. The energetic performance of this elementary motor is also evaluated.

In section 5.4, a pneumatic motor with enhanced efficiency will be described where an additional expansion chamber is coupled to the first single piston system. Pressures, forces and torques will be simulated. Finally the energetic performance will be calculated and compared to the performance of the single cylinder system. An experimental system is also realized.

2. OPERATING PRINCIPLE OF THE MOTOR WITHOUT EXPANSION

A double acting linear cylinder is used as prime mover Fig. (**5.1**). The two working chambers are filled alternatively with compressed air. Each filling stroke is characterized by its displacement work with a constant force exerted by the piston. Before entering the reversal motion, where the force is then exerted by the opposite chamber, the air of the first chamber is released to the ambient, losing its energetic content bound to the pressure. The corresponding energy loss factor or equivalent efficiency has been presented in Chapter 2.3.2.

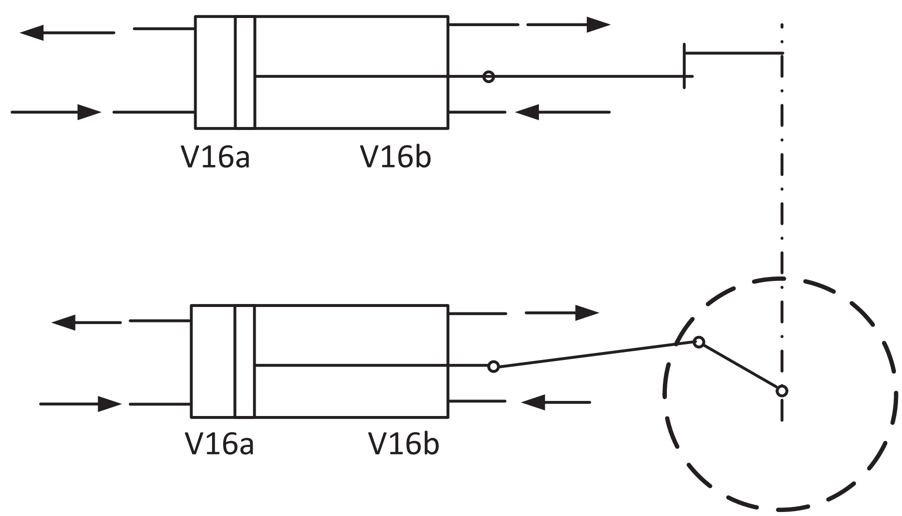

Fig. (5.1). Reciprocating cylinder with crankshaft and connecting rod (top view and side view).

2.1. Mathematical Description of the Piston/Crankshaft Assembly

In Fig. (**5.2**), the piston is represented with the connecting rod and the crankshaft. The parameters are indicated as r, the radius of the crankshaft, l the length of the connecting rod, and Φ the angle of rotation of the crankshaft. The diameter of the piston, d and its position x are also indicated.

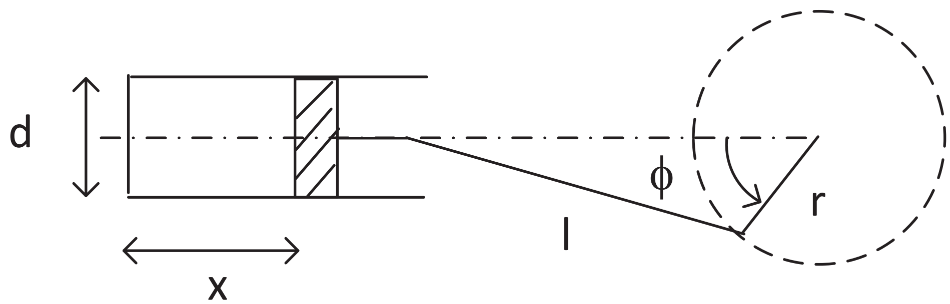

Fig. (5.2). Piston, crankshaft and connecting rod.

The position of the piston is given through rel. (5.1):

$$x = r(1-\cos\varphi) + \frac{\lambda}{2} r \sin^2\varphi \quad \textbf{(5.1)}$$

Where the connecting rod ratio λ is used and is defined as:

$$\lambda = \frac{r}{l} \quad \textbf{(5.2)}$$

The velocity of the piston is given by:

$$v = \omega \cdot r \cdot \sin\varphi(1+\lambda\cos\varphi) \quad \textbf{(5.3)}$$

In the simulation process, the torque developed by the motor is calculated through the indirect calculation of the power.

If the force exerted on the piston is given by:

$$F_p = p \cdot A \quad \textbf{(5.4)}$$

CHAPTER 6

Linear Pneumatic Cylinder Assembly with Reduced Air Consumption

Abstract: The method of adding expansion work to pneumatic actuators is studied for classical linear cylinders. The operating principle of new cylinder assemblies is presented. A first simulation set illustrates the performance of the new assembly and tries to define the parameters of a single cylinder which produces the same mechanical performance. Acceleration, speed and reached position within a given time are the conditions for the comparison. Then, the air consumption of both compared systems is calculated. With an experimental set-up, a parasitic effect is observed, which consists of a pre-expansion transient due to parasitic dead volumes related to the tubing and internal volumes of the valves. A second assembly is realized with larger volumetry in order to observe the dependency of the parasitic effect from the size of the cylinders. For the control, a system with simpler control valves is also studied.

Keywords: Adiabatic expansion, Cylinder assemblies, Dead volumes, Energetic performance, Energy efficiency, Expansion work, Linear cylinders.

1. INRODUCTION

In Chapters 3, 4 and 5, different systems using pneumatic actuators have been analysed and discussed, especially from the point of view of energetic efficiency. The principle of combining two types of production of mechanical work has been applied, namely the production of so-called displacement work under constant pressure and expansion work realized through the variation of the active volume. The production of this expansion work has been realized with air transfer from a first volume to a second one of larger dimensions, or simply by controlling the intake valve of one volume.

1.1. New Cylinder Assemblies

In the present chapter, the same principle of recovering the thermodynamic content of the pressurized air is applied to linear cylinders, where in addition to the simple displacement work produced in a conventional cylinder, the air is addi-

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tionally expanded in a supplementary pneumatic chamber system, allowing to recover a significant part of the injected enthalpy [25].

Fig. (**6.1**) shows one of the possible arrangements of the new assembly, where the displacement work is produced by a central cylinder and where the expansion of the air is done within three peripheral cylinders mechanically coupled to the central one. In such an arrangement, the volumetric ratio of the expansion is equal to 3.

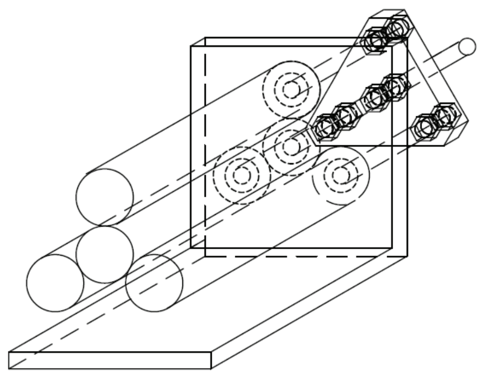

Fig. (6.1). One of the possible arrangements of the proposed cylinder assembly.

The system illustrated in Fig. (**6.1**) is studied, and its performance is compared with a single cylinder producing the same work. Fig. (**6.2**) gives a front view and a side view of the proposed system, while Fig. (**6.2b**) shows the single cylinder with compatible mechanical interface elements.

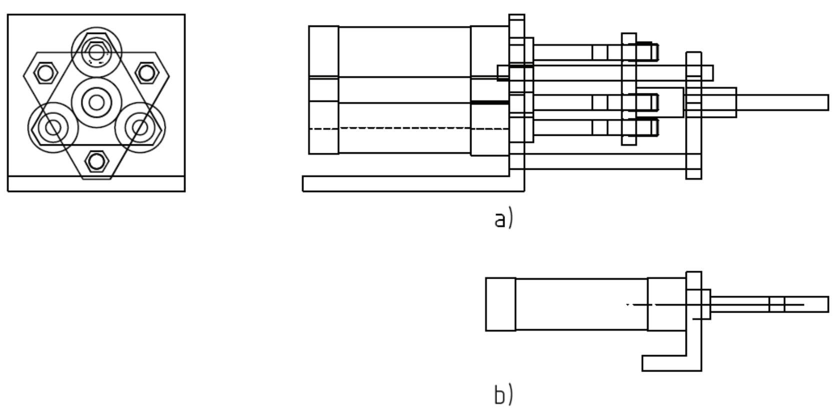

Fig. (6.2). Proposed system a) Front and side view, b) single cylinder with a compatible interface.

The cylinder assembly with one central and three peripheral cylinders is represented in Fig. (**6.2**) is only one of the possibilities of coupling cylinders for realizing the additional expansion work. Three other solutions are represented in Fig. (**6.3**).

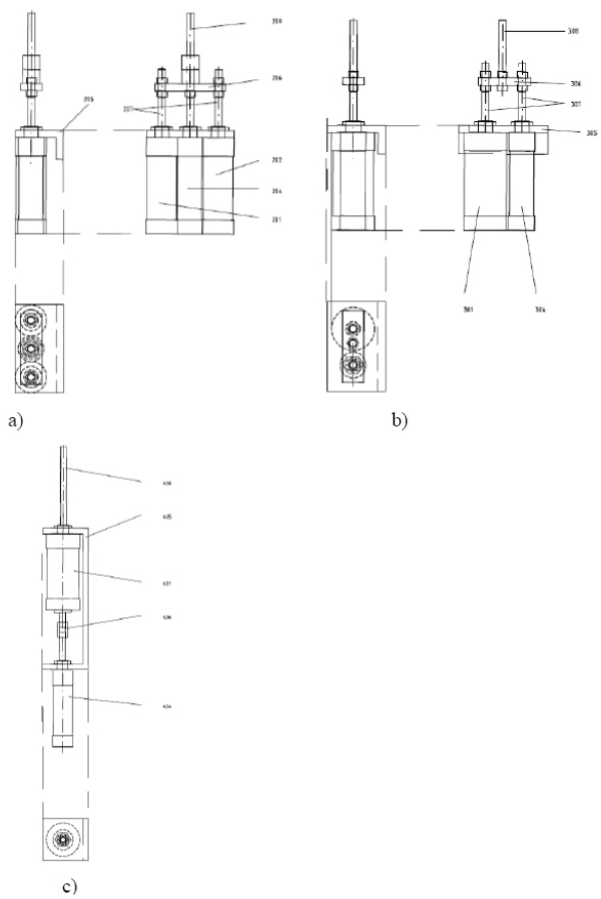

Fig. (6.3). Different configurations of coupled cylinder-assemblies.

In Fig. (**6.3a**), central cylinder is used for the displacement work and two lateral cylinders assume the function of producing expansion work. In Fig. (**6.5**), the small cylinder produces the displacement work while the parallel running larger cylinder produces the additional expansion work.

In Fig. (**6.3**), a similar assembly is shown, but where the two cylinders are placed on the same axis. In this configuration, no torsional effort is produced. One of the two coaxially running cylinders must be of the double rod type.

CHAPTER 7

The Effect of the Dead Volumes and Pre-Expansion on the Produced Work

Abstract: In Chapter 6, the parasitic effect of pre-expansion due to the presence of dead volumes has been observed. This effect will now be analyzed in more detail in this chapter. Especially the influence of this pre-expansion on the total produced mechanical work is calculated. Different pre-expansion factors are considered and the mechanical work of an ideal system without pre-expansion will be compared with the reduced mechanical work of a cylinder assembly affected by pre-expansion.

Keywords: Dead Volumes, Pre-expansion, Pre-expansion Factor, Torque Reduction.

1. INTRODUCTION

The experimental results realized with linear cylinders and described in Chapter 6 have revealed the parasitic effect of the pre-expansion due to the presence of dead volumes. This effect is inherent to all systems where pressured air is intended to be transferred from one volume to another. The systems described in Chapters 4, 5 and 6 have been simulated in ideal conditions where no dead volumes have been considered. Typically, the evolution of the pressure during the expansion phase, as illustrated in Fig. (**4.4**), shows that the initial value of pressure P_2 of the expansion process corresponds to the value of the intake pressure P_{in} of the air in the small chamber established during the previous stroke. The absence of discontinuity in the pressure by the changeover from filling to expanding is due to the condition that the interconnection of the volumes V_{1b} and V_{2a} results in $V_{1b}+V_{2a,}$ as represented in Fig. (**4.3b**). In practice, the interconnection of the volumes must be modelled taking into account the dead volumes as represented in Fig. (**6.14**) and (**6.17**).

The pre-expansion phenomenon observed with the small cylinders in the previous chapter can be characterized by the ratio of the pressure after the opening of the transfer valve to the initial value of the pressure upstream of this valve before its opening (P1 to P1' in Fig. (**6.18**)). This ratio of the pressure discontinuity is called the pre-expansion factor.

In the next sections, the pre-expansion and its effect on the developed torque will be simulated. The considered system corresponds to the system described in Chapter 4, namely the system with coupled semi-rotary actuators. But the evolution of the volumes of the complementary smaller and larger chambers of this system is identical to the evolution of the coupled chambers of the systems using linear cylinders.

First, the discontinuity of the pressure will be shown in Section 7.1.1 In this simulation, the intake pressure and the parameters of the devices of the system of Chapter 4 are considered. This will allow to make a comparison of an ideal system with a system with dead volumes. The comparison will be done on the base of the different torques produced by the ideal and non-ideal system models, and for two different values of the pre-expansion factor. A pre-expansion factor of 0.6 characterises the system simulated in Section 7.1.2 and a value of 0.8 characterises the system simulated in Section 7.1.3.

1.1. Discontinuity of the Pressure

The discontinuity of the pressure illustrating the pre-expansion due to dead volumes is represented in Fig. (**7.1**). The initial (filling) pressure is 10 bar, and the pre-expansion factor is equal to 0.6 The expansion process is conditioned by the same parameters as for the example of Chapter 4.

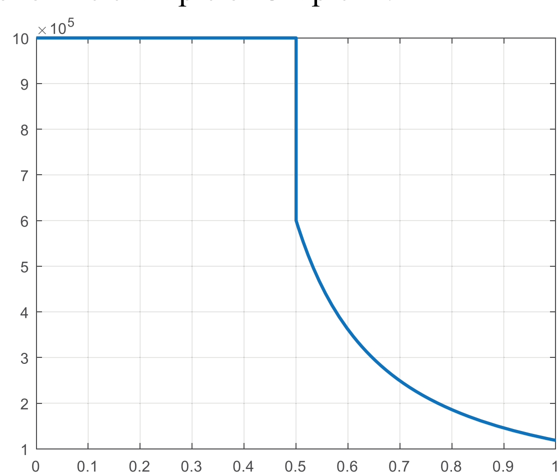

Fig (7.1). Pressure discontinuity due to pre-expansion X: Time (s) (similar to Fig. 4.3 to 4.9) Y: Pressure (bar).

1.2. Torques Developed with a Pre-Expansion Factor of 0.6

Fig. (**7.2**) illustrates the torque contributions of both sides of the wing of the first (small) actuator during the second half period. The curves of an ideal system without pre-expansion are represented for comparison with the curves of a system with pre-expansion. The curves of the system without pre-expansion are identic to the curves simulated in Chapter 4.

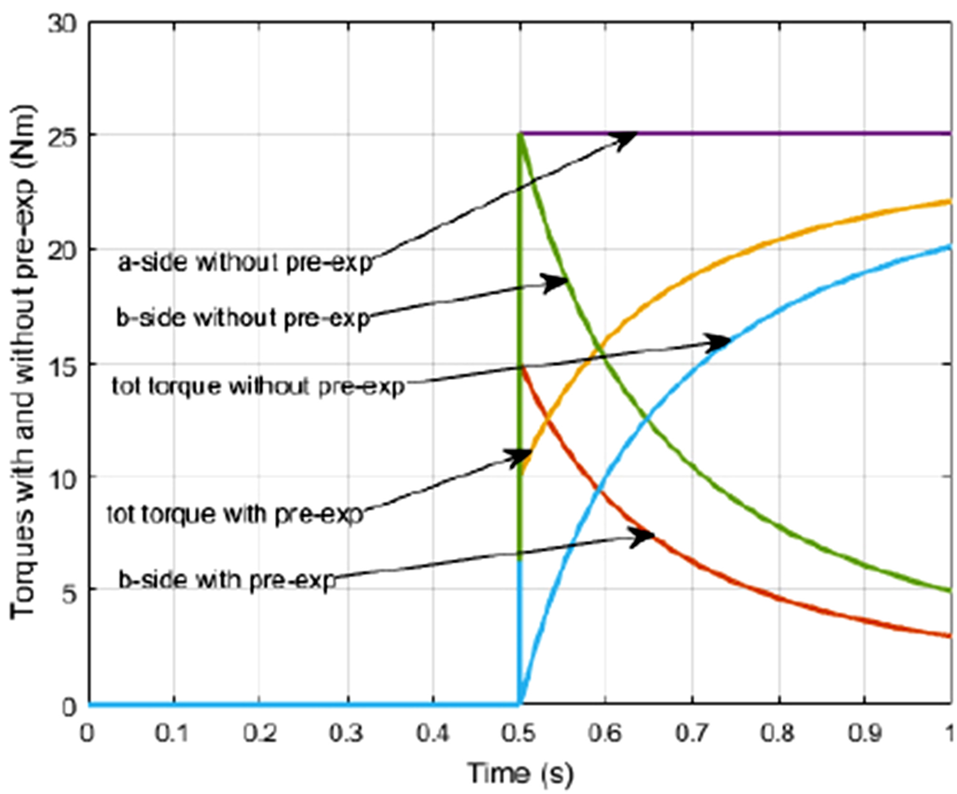

Fig. (7.2). Torque contributions of the first (small) actuator (Nm).

In Fig. (**7.3**), the torque contributions of the first (smaller) and second (larger) actuators are represented. The curves show the phenomenon in the second half-period. The curves of a system with and without pre-expansion are represented.

CHAPTER 8

Application Example: A Pneumatic Driven Hydrogen Compressor with Increased Efficiency

Abstract: In this chapter, an application example is studied where the energetic efficiency of a pneumatically driven device is of importance. The chosen example consists of an air-driven gas booster used as a Hydrogen compressor in a refuel station for H_2 driven cars. The needed force for the driving of the compression cylinders is calculated, and a new pneumatic motor based on the principle of adding expansion work is proposed. The new motor is designed for sufficient effort for moving the mobile equipment under the maximum compression force.

The air consumption of the new system is calculated, and finally, the air savings in comparison to a classical air-driven booster. The simulation is completed with a dynamic part showing the dynamic performance in terms of velocity and time to reach the final position of the pistons

Keywords: Air-driven gas-booster, Design, Dynamic simulation, Energetic efficiency, Saving of air.

1. INTRODUCTION

In many domains of gas pressurization, but specifically for the refueling stations for hydrogen powered vehicles, so-called gas-boosters are used. These gas-boosters are driven by a central double acting pneumatic motor. In this chapter, an air driven hydrogen compressor is studied, and a new approach is presented based on the principles described in Chapters 4 and 5. This can significantly improve the energy efficiency of the compression stations, or in other terms, can significantly reduce the consumption of compressed air for an identic output performance.

A schematic representation of a hydrogen refilling station is given in Fig. (**8.1**) where hydrogen is produced from an electrolyser, fed from photovoltaic panels. The high-pressure compression stage based on a gas-booster is also represented. In this example, the air compressor, which provides the activation fluid of the gas booster, is also powered by renewable power sources.

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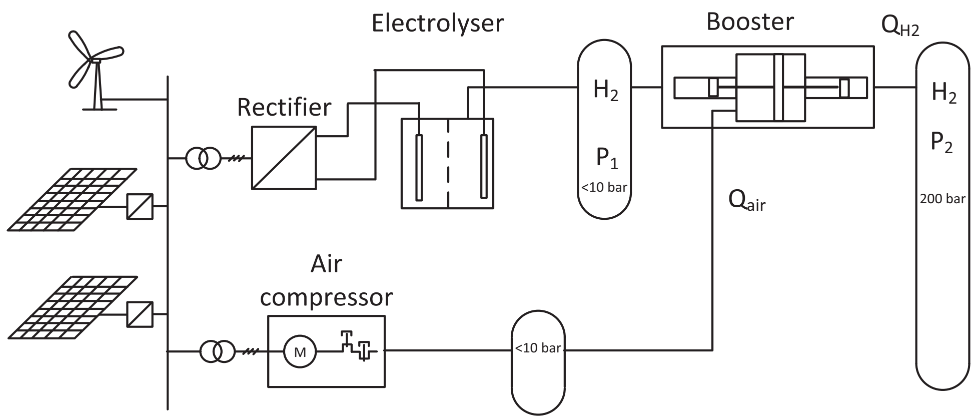

Fig. (8.1). Hydrogen refueling station using a gas-booster compression stage.

The classical gas boosters with its reciprocating equipment are represented schematically in Fig. (**8.2**). It is composed of two high-pressure gas compression cylinders at both ends, driven by a central larger double-effect pneumatic actuator. As with all usual pneumatic actuators, this drive presents a low energetic performance in the sense that the internal content of pressurized air in the cylinder is simply released to the surrounding at the end of the strokes, before initiating the return motion of the system, losing a large part of the injected enthalpy.

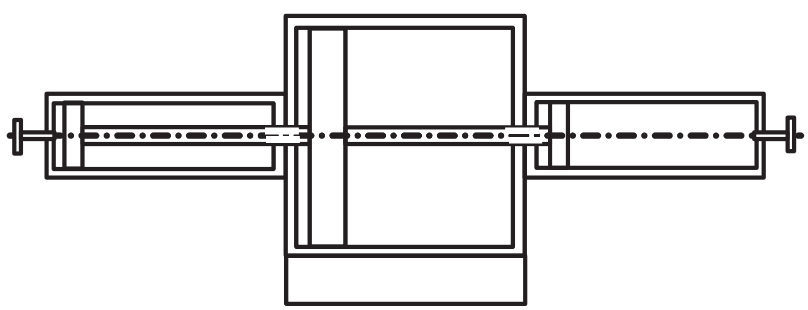

Fig. (8.2). Example of a classical air driven gas booster.

In the new proposed system, the volume of the original cylinder is reduced and a second larger cylinder is added to the system in order to recover a great part of the compressed air energy by thermodynamic expansion.

2. DATA AND PERFORMANCE OF THE ORIGINAL BOOSTER

An original booster system is first considered, whose data are summarized in Table **8.1**. This system comprises a pneumatic motor cylinder driving two single compression pistons (Fig. **8.2**). The pneumatic motor is operated conventionally under constant pressure. The compression stage produces the output pressure up to a pressure of 160 bar, where the exhaust valve opens.

Table 8.1. Characteristics of the original booster DLE 15.

Diameter of the air piston	$\mathbf{D_{piston}}$	**153 mm**
Diameter of the rod	D_{rod}	16.6 mm
Length of the stroke	l_{stroke}	94.5 mm
Diameter of the compressor piston	D_{conpr}	28.16 mm
Surface of the air piston	A_{piston}	18385 mm^2
Surface of the rod	A_{rod}	216 mm^2
Active surface of the air piston	A_0	18169 mm^2
Active surface of the compression piston	A_{compr}	620 mm^2
Volume of the compressor chamber	V_{compr}	58604 mm^3
Inlet pressure of air	P_{in_air}	8 bar
Inlet pressure of the gas	P_{in_gas}	15 bar
Output pressure of the gas	P_{out_gas}	160 bar

The constant force exerted by the pneumatic motor is:

$$F_{pneum\_0} = P_{in\_air} \cdot A_0 = (8e^5 - 1e^5)\frac{N}{m^2} \cdot 0.018170 m^2 = 12719 N \qquad \textbf{(8.1)}$$

At the end of the compression, when the exhaust valve opens, the counter force of the compression piston is:

$$F_{gas} = P_{\text{out\_gas}} \cdot A_{compr} = (160e^5 - 1e^5)\frac{N}{m^2} \cdot 0.000620 m^2 = 9858 N \qquad \textbf{(8.2)}$$

The evolution of the constant driving force and the load force of one compression cylinder is represented in Fig. (**8.3**). The compression force is calculated with pressure varying according to an adiabatic compression process as:

CHAPTER 9

Conclusion

Abstract: The chapter serves as a conclusion to the different principles, systems and application examples described in this book.

From the original air-powered diving lamp driven by a semi-rotary pneumatic actuator to the final example of the gas booster, different systems have been proposed with a large benefit in terms of energetic efficiency or in other terms, in the reduced amount of air consumed.

Keywords: Air-savings, Energetic performance, Linear actuators, Pneumatic actuators, System assemblies, Semi-rotary actuators.

CONCLUSION

Original development of a compressed air-fed diving lamp has been the trigger of investigations on the energetic efficiency of pneumatic devices. This system operates as a compressed air energy storage system, where the storage pressure is at a high level. Reflexions on the possibility of extracting the whole content of the supplied air to the conversion devices have been done.

Two different phenomena in the operation of classical pneumatic devices in the diving lamp example are the cause of poor energetic efficiency. The first phenomenon is in relation to the use of a pressure release valve placed between the high-pressure reservoir and the converting device, which operates at low pressure. Such a reduction valve is responsible for a significant energy loss called exergy loss.

The context of energy storage by compressed air calls for not using a pressure reducing valve. As an alternative, it is more judicious for reasons of energy efficiency to store the air at a pressure level close to the pressure level of the conversion devices. In this sense, the solution of storing energy in air balloons submerged at reasonable depths which is proposed by the University of Nottingham, makes it possible to have a practically zero pressure difference between the storage tank and the conversion device and to renounce the reducing

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valve. An immersion depth of 100 m imposes a pressure of 10 bar. Additionally, inflating and deflating the balloons at a constant depth has the advantage to realize the storage at constant pressure, independent of the state of energy of the storage system.

The second reason for the poor resulting efficiency is the operation principle of classical pneumatic actuators, where after the filling of the active volume with air under pressure, this air is simply released to the surrounding, without recovering its thermodynamic content.

As a result, the proposal is made to add to an original pneumatic actuator a parallel running expansion device of the same type but with a larger active volume. The basic mechanism is then to realize a pneumatic to mechanic energy conversion where not only a so-called displacement work under constant pressure is produced but where an additional expansion work is added through the transfer of the air from the closing chambers of the small volume to the opening ones of the larger one, allowing to significantly increase the energetic performance of the system for an identical consumption of air.

The main contribution of this book is given in Chapter 4, where the principle of coupling two semi-rotary actuators is analysed. The oscillating angular displacement of these actuators is transformed into a unidirectional rotation through a so-called motion rectifier. The evolution of the volumes of the cascaded actuators is simulated, with the related expansion and variation of the pressure. The developed torques of the individual devices as the global torque are simulated for an evaluation of the energetic performance of the proposed system. Adiabatic and isothermal expansions have been simulated, indicating that the performance of a pneumatic device can be increased by nearly 80% for an adiabatic expansion, or over 100% if the expansion is done in isothermal conditions.

Then, an alternative system where the displacement work and the expansion are realized in one single device is studied. The principle of operation consists of filling the chambers of the device during a part of the angular displacement, and then, after closing the intake valve, to realise the expansion in the remaining volume of the stroke. For this system, the different torques and produced mechanic work are analysed, together with the energetic performance. The main outcome is that the energetic performance is identical (average value of the torque) to the performance of the previous system, but the developed torque presents a higher variation over one stroke.

The two systems with coupled and single actuators have been realized in the form of small demonstrators. Their dedicated control system and valves with open-loop and closed-loop with position sensors of the actuator's rotor have been verified.

Some complementary reflexions analyse the possibility to obtain a reversible power flow of the conversion with semi-rotating actuators. The reversibility of the power has its motivation in the domain of energy storage, where compression and expansion should be realized with the same machine.

Chapter 5 of the book is dedicated to the realization of a pneumatic motor based on linear actuators. The same principle of adding constant pressure displacement work and expansion work is applied to this motor. The energetic performance of this motor is obtained as identic to the previously analysed systems, even if the evolution of the pistons is non-linear due to the use of a crankshaft and its related typical dynamics.

The same motor is then analysed where constant pressure displacement work and expansion are realized in the same cylinder.

Chapter 6 is dedicated to the operation of linear pneumatic cylinders, where the same principle of adding an expansion chamber is applied. Different configurations are described, which have been explained in a patent application. A first configuration is analysed and has been realized where the constant pressure work is done in a central cylinder and where the expansion is realized within three identic cylinders placed around the first one. All four piston rods are mechanically interconnected.

A second configuration is presented where one central and two lateral cylinders are coupled.

Through the experimentation of these systems, a parasitic effect was observed, namely the effect of the presence of dead volumes. These dead volumes represent the volumes of the valves and of the connecting tubes. Especially the opening of the transfer valve at the beginning of the expansion causes a pressure discontinuity which can be modelled as a pre-expansion.

Chapter 7 analyses the effect of the dead volumes and of the related pre-expansion on the energetic efficiency of the proposed assemblies. The mechanical performance is analysed by simulation for different values of the pre-expansion factor, assimilated to the importance of the dead volumes in regard to the active volumes of the devices.

In the Eighth chapter of the book, a specific application example is given. This example concerns the so-called air driven gas boosters utilized as compression stages in refill stations for hydrogen powered automotive vehicles.

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APPENDIX 1

A1. ENERGY CONTENT OF AN AIR RESERVOIR

A1.1. Description of the System

The considered system comprises one air reservoir of a volume V_1, connected to a cylinder/piston system, as represented in Fig. (**6.34**). The pressure level at t_0 is equal to P_1. The system is idealised as a piston/cylinder of infinite length. The section of the piston is defined as A_p.

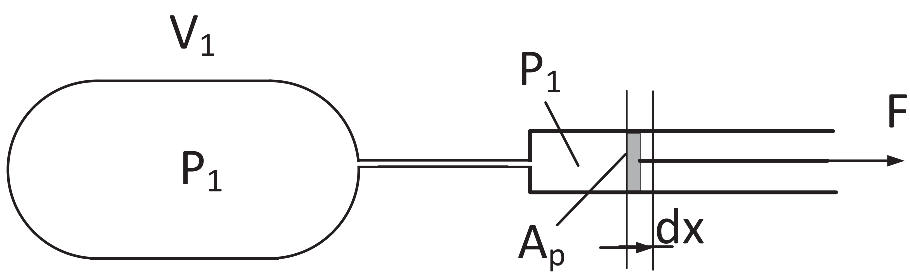

Fig. (A1.1). Idealized vessel and infinite cylinder.

For such a system, the following hypotheses are made:

Isothermal expansion: In both the reservoir and the piston system, there is a decrease in the pressure and a corresponding decrease in the temperature in relation to the ideal gas law. It is assumed that temperature remains constant during the entire expansion process, since it is a sufficiently slow expansion (quasi equilibrium), allowing constant compensation of the temperature decrease by heat transfer from the surrounding.

Infinite cylinder: Particularly for the high pressure cylinders, the volume variation of a cylinder of real dimensions only allows a limited pressure decrease for one stroke. In practice, the principle of successive expansions and fillings is used, which includes additional dynamic phenomena of pressure exchange and gas circulation into and out of the cylinder of finite volume. In order to evaluate the ideal potential of the pressurized reservoir, it is considered that the expansion is made in a single cylinder of a given section and of infinite length, allowing to make expansion of the air in a high pressure ratio.

Friction: The losses due to mechanical friction are neglected, as well as the air friction (viscous dissipation) inside of the different air ducts.

A1.2. Mechanical Work by Expansion

Goal: Calculation of the mechanical work to be produced with the piston/cylinder system when the pressure is decreasing from P_1 to the atmospheric pressure.

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The calculation should use the following equation:

$$W_{piston} = \int_0^{Xat} F(x)\cdot dx = \int_0^{Xat} (P_{res} - P_{at})\cdot A_p \cdot dx = \int_0^{Vat} (P_{res} - P_{at})\cdot dv \qquad \text{(A1.1)}$$

The expansion volume considers the presence of an initial volume V_1 (volume of the reservoir) in which the pressure is initially P_1. The displacement of the piston leads to an increase of the volume towards a new value V_{atm}, to be reached when the internal pressure reaches the atmospheric pressure P_{atm}. This volume can be easily calculated for an isothermal process:

$$V_{atm} = \frac{V_1 \cdot P_1}{P_{atm}} \qquad \textbf{(A1.2)}$$

The mechanical work produced by the piston force can be calculated through:

$$W_{piston} = \int_{V_1}^{Vatm} (P_{res} - P_{atm})\cdot dv = \int_{V_1}^{Vatm} \left(\left(\frac{P_1 \cdot V_1}{v}\right) - P_{atm}\right)\cdot dv$$

$$W_{piston} = \int_{V_1}^{Vatm} \left(\left(\frac{P_1 \cdot V_1}{v}\right)\cdot dv - \int_{V_1}^{Vatm} P_{atm}\cdot dv = \right. \qquad \textbf{(A1.3)}$$

$$= \int_{V_1}^{Vatm} \left(\left(\frac{P_1 \cdot V_1}{v}\right)\cdot dv - P_{atm}\cdot V_{atm} + P_{atm}\cdot V_1 = \right.$$

$$= P_1 \cdot V_1 \cdot \ln\frac{V_{atm}}{V_1} - P_1 \cdot V_1 + \frac{P_{atm}}{P_1} P_1 \cdot V_1 \qquad \textbf{(A1.4)}$$

$$W_{piston} = P_1 \cdot V_1 \cdot \left(\ln\frac{P_1}{P_{atm}} - 1 + \frac{P_{atm}}{P_1}\right) \qquad \textbf{(A1.5)}$$

Let us call this expression Lemofouet’s formula.

In opposition to this expression, the common formulas give different results, mainly if the ratio of the pressures is low.

In [2], the relation gives:

« Necessary work for isothermal compression »

$$W_i = \int_{p_1}^{p_2} mRT_1 \frac{dp}{p} = mRT_1 \cdot \ln \frac{p_2}{p_1}, \quad \text{with} \quad mRT_1 = P_1V_1 \qquad \textbf{(A1.6)}$$

This relation can be used for expansion and corresponds to the first term of rel. (A1.5). It does not consider the effect of the external pressure on the rear side of the piston. In fact, this relation gives the maximum of work that can be extracted from the reservoir by expansion from a pressure P_2 to a pressure P_1 under isothermal conditions, corresponding to a situation where the external pressure is zero.

Comparison of the results

One reservoir of 200 liters at 300 bar contents according to the expression (A1.6):

$$W_i = 0.2*300*10^5 \ln(300) = 34{,}2 \text{ MJ} \qquad \textbf{(A1.7)}$$

According the Lemofouet’s formula (A1.5), it is possible (at atmospheric pressure) to extract

$W_{piston} = 0.2*300*10^5 (1/300 – 1 + \ln(300)) = 28{,}24$ MJ

For the same volume of 200 liters, the storage at 8 bar gives:

According (A1.5): $W_{piston} = 192$ kJ

According (A1.6): $W_i = 332$ kJ

APPENDIX 2

A2. MECHANICAL FORCES AND ENERGETIC PROPERTIES OF THE 100 MM LINEAR CYLINDER ASSEMBLY

A2.1. Introduction

In Chapter 6, linear cylinder assemblies are presented where the principle of the "added expansion" has been applied. The numeric simulations have shown the dynamic behaviour of a specific assembly using a set of 1+3 identical cylinders with short strokes (20mm). The related experimental investigations have shown the non-insignificant effect of the dead volumes. Then a second assembly with 1+2 cylinders and a longer stroke (100mm) was realized where the effect of the dead volumes was supposed to be smaller.

In this appendix, the quasi-static behaviour of the new assembly is analyzed (Fig. **6.15** and Table **6.2**), as well as the energetic properties. The comparison is made with a single cylinder operated in a classical way, without expansion, which consumes the same amount of compressed air as the new assembly.

A2.2. Quasi-Static Behavior of the new Assembly

The simulation of the 1+2 cylinder-assembly is done in a similar way as the simulation of the semi-rotary vane-type actuators in Chapter 4, namely a system where the piston's movement is imposed by an external source. This allows to represent the value of the generated forces in dependency of the piston's position.

The simulated curves represent the pressure (Fig. **A2.1**) and the forces exerted on the sides of both pistons, from the previously calculated pressure. This pressure P_{exp} is calculated in function of the position of the pistons, as a function of the ratio of the volumes of both cylinders, according to the rule of an adiabatic expansion.

$$P_{\exp} = P_{in\_\mathrm{air}} \left(\frac{V_{1\max}}{V_{1a} + V_{2b}} \right)^{\gamma} \quad \text{with g = 1.4} \tag{A2.1}$$

For the example shown in Fig. (**6.15**), a 12 mm cylinder is used as the filling device, and two 16 mm cylinders are the expansion device. With a stroke of 100 mm, the volumes of the first and second cylinders become:

$$V_{1\max} = \frac{(0.012m)^2 \cdot \pi}{4} \cdot 0.1m = 0.0000113m^3 = 11.3cm^3 \tag{A2.2}$$

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$$V_{2\max} = 2 \cdot \frac{(0.016m)^2 \cdot \pi}{4} \cdot 0.1m = 0.0000402m^3 = 40.2cm^3 \quad \textbf{(A2.3)}$$

The volumetric ration is then:

$$R_{vol} = \frac{V_{1\max}}{V_{2\max}} = 0.281 = \frac{1}{3.55} \quad \textbf{(A2.4)}$$

During the movement of the pistons from left to right, the volume V_{1a} varies from V_{1max} to zero, and the volume V_{2b} varies from zero to V_{2max}. At the end of the stroke, the volumetric ratio is V_{1max}/V_{2max}. With a choice of 1/3.55 for this ratio, the pressure ratio becomes:

$$\frac{P_{\exp}}{P_{in\_\mathrm{air}}} = \left(\frac{1}{3.55}\right)^{1.4} = 0.169 \quad \textbf{(A2.5)}$$

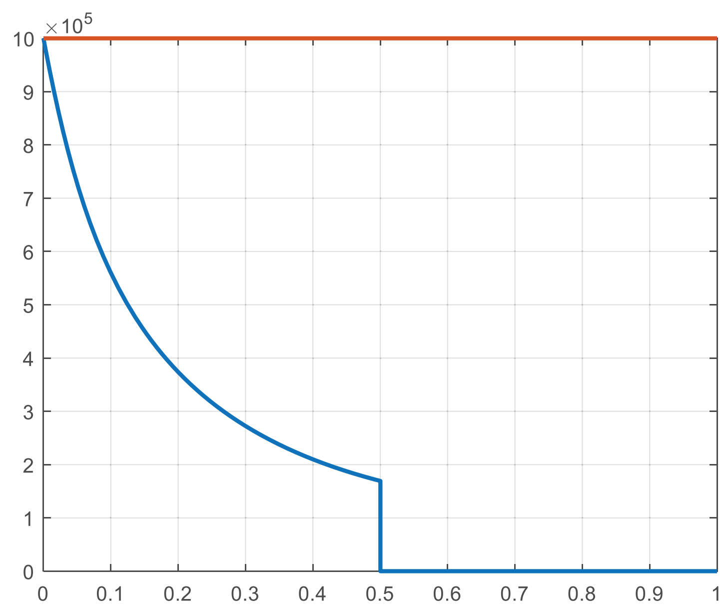

Fig. (A2.1). Pressure by expansion (N/m^2). Time (s).

For the diagram of Fig. (**A2.1**), an input pressure of 10 bar was chosen.

The curves in Fig. (**A2.2**) show the evolution of the forces on the first piston. The red curve corresponds to the constant force related to the intake at constant pressure. The value of this force is calculated through:

$$F_{1a} = P_{in\_air} \cdot A_1 \tag{A2.6}$$

where A_1 is the active surface of the first (small) piston.

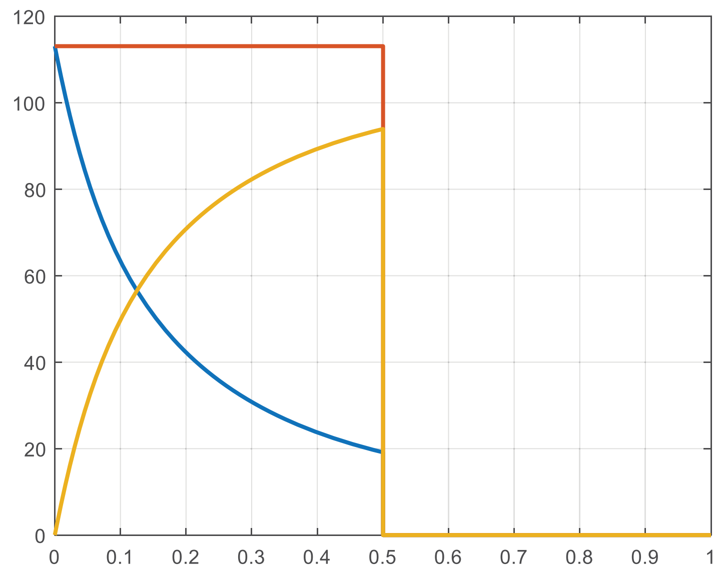

Fig. (A2.2). Forces exerted on the small piston (left-to-right stroke) (N) Time (s).

The blue curve corresponds to the force on the opposite side of this piston due to the pressure during the expansion. This force is given by:

$$F_{1b} = P_{\exp} \cdot A_1 \tag{A2.7}$$

The yellow curve in Fig. (**A2.2**) corresponds to the global force produced by the first cylinder, namely:

$$F_{1glob} = F_{1a} - F_{1b} \tag{A2.8}$$

In Fig. (**A2.3**), the force contributions of the second cylinder are represented. The red curve is the global force exerted by the second piston F_{2a}. This force takes the value of:

$$F_{2glob} = F_{2a} - F_{2b} = P_{\exp} \cdot A_2 - P_{atm} \cdot A_2 \tag{A2.9}$$

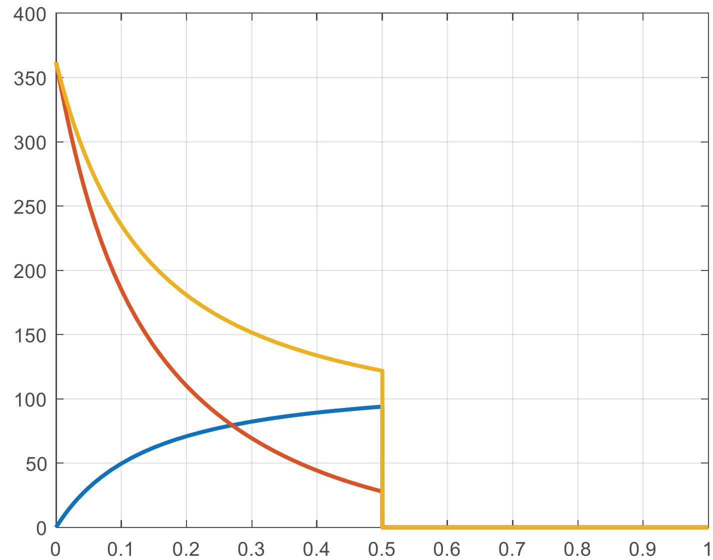

Fig. (A2.3). Global forces of the two pistons and total force of the new assembly (N), Time (s).

The second term in rel. (12) illustrates the fact that on the opposite side of this piston, the exhaust valve is open, and the atmospheric pressure must be taken into account.

In the same Fig. (**8**), the global force contribution of the first cylinder is represented again (blue curve), as the total force produced by the new assembly (yellow). This total force is

$$F_{tot} = F_{1glob} + F_{2glob} \tag{A2.10}$$

Finally, Fig. (**A2.4**) shows the comparison of the performance of the new system with the performance of the single (small) cylinder operated alone, without expansion. The air consumption of both systems is the same. The blue curve shows the force produced by the new system and the yellow one gives its average value. The average value of the produced force is:

$$F_{ave} = \frac{1}{T/2}\int_0^{T/2} F_{tot} = 185N \tag{A2.11}$$

The force produced by the single cylinder without expansion is given by:

$$F_{sgle} = (P_{in\_air} - P_{atm}) \cdot A_1 = 9e5N/m^2 \cdot 0.000113m^2 = 101.7N \tag{A2.12}$$

and takes into account that the opposite side of the piston receives the atmospheric pressure. A_1 is the area of the small piston (12mm).

On the basis of this comparison, the new cylinder assembly produces an average force equal to:

$$F_{ave} = \frac{185N}{101.7N} F_{sgle} = 182\% \cdot F_{sgle} \quad \textbf{(A2.13)}$$

Another advantage of the new system is that it can accelerate the mass of a given application with an important force at the beginning of the motion. The maximum force of the new system being

$$F_{\max} = (P_{in\_air} - P_{atm}) \cdot A_2 = 9e5N / m^2 \cdot 0.000402m^2 = 361.9N \quad \textbf{(A2.14)}$$

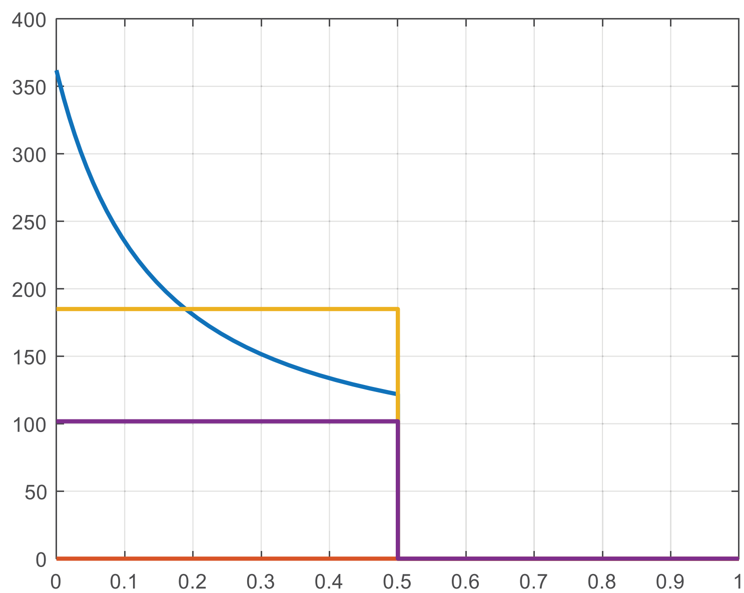

Fig. (A2.4). Total force with average and force of the small cylinder alone (without expansion) (N), Time (s).

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Alfred Rufer

Prof. Alfred Rufer received a Master's degree from the Swiss Federal Institute of Technology in Lausanne (EPFL), Switzerland, in 1976. From 1976 to 1978, he was a research assistant with EPFL's Chair of Industrial Electronics (Prof. H. Bühler). In 1978, Alfred Rufer joined ABB, Turgi, Switzerland, where he worked in the fields of power electronics and control for high-power variable frequency converters for drives. He has participated in several development teams on new applications of power electronics, like renewable energies. In 1985, he was a group leader for power electronics development at ABB. In 1993, he became an assistant professor at EPFL. In 1996, he was elected as a full professor and head of the Industrial Electronics Laboratory at EPFL. The Industrial Electronics Lab (LEI) is active in power electronics used in energy conversion and energy storage, in modelling and simulation of systems, including control strategies and control circuits. He has several patents and is the author of publications on power electronics and applications. In 2006, he was elected to the IEEE fellow grade. In 2016, Alfred Rufer retired, and got the title of professor emeritus. His research is dedicated to energy efficiency and pneumatic actuators. He has published different quantitative analyses in the field of energy.