J. Webster (ed.), *Wiley Encyclopedia of Electrical and Electronics Engineering*

COMPRESSED AIR ENERGY STORAGE

Energy Conversion and Storage Systems

Energy conversion paths are either direct or indirect (Fig. 1). With indirect technologies the line of energy conversion is the following: the primary source is exothermally oxidized (burnt), the chemical energy being converted into thermal energy. By a thermal agent (motor fluid), the thermal energy is used as such, or by using a thermal machine (turbine, engine) it is converted into mechanical energy actuating an energy-generating working engine or generator. With direct technologies thermal intermediation is eliminated along with its thermodynamic imperfections, and electrical energy is directly gained, for instance, from the chemical energy of fuel. Wind energy falls into this category.

There are several competitive possibilities for converting primary energy into electrical energy: hydraulic, wind-driven, electrochemical, and nuclear (including fusion); solar energy may be directly utilized on a photovoltaic line.

Of all primary energy forms suitable for sustainable development, wind energy is the most advantageous: it is practically inexhaustible, nonpolluting, available everywhere, and free of charge, since no primary extraction is required. In converting the wind energy to other forms, the following stages are to be noted: extraction, conversion, storage, and consumption. For wind, energy storage is necessary both because of the limitations of wind (it is irregularly distributed in time and space, and it is dilute, that is, it has low concentration per unit area), and because of the variety of its consumers. Figure 2 presents a general diagram of extraction, conversion, storage, and utilization of wind energy.

In the first stage (extracting energy from wind), mechanical energy is gained as rotation, translation, or oscillation, depending on the type of the extraction installation. This motion may be used in the second stage for conversion into another form of energy by liquid pumping, gas compression, or electrical or heat generation. The energy gained may be stored by using pneumatic or hydraulic systems, electric batteries, heat, hydrogen, flywheels, etc. This is the third stage. The fourth stage, that of consumption, consists in supplying energy to consumers in the form of heat, electricity, water, etc.

A comparison between different energy storage systems is presented in Fig. 3. For compressed air, starting with the data from the literature, which takes into consideration only the mechanical component of the energy, we have also taken into consideration the possibilities of utilizing the thermal component.

Means of Energy Storage by Compressed Air

Energy storage by compressed air can be accomplished using storage tanks, hydropneumatic accumulators, and pneumatic network elements.

Compressed Air Storage Using Storage Tanks. Figure 4 shows the diagram of compressed air storage. The installation is made up of the following elements: wind turbine 1, hydrostatic transmission 2,

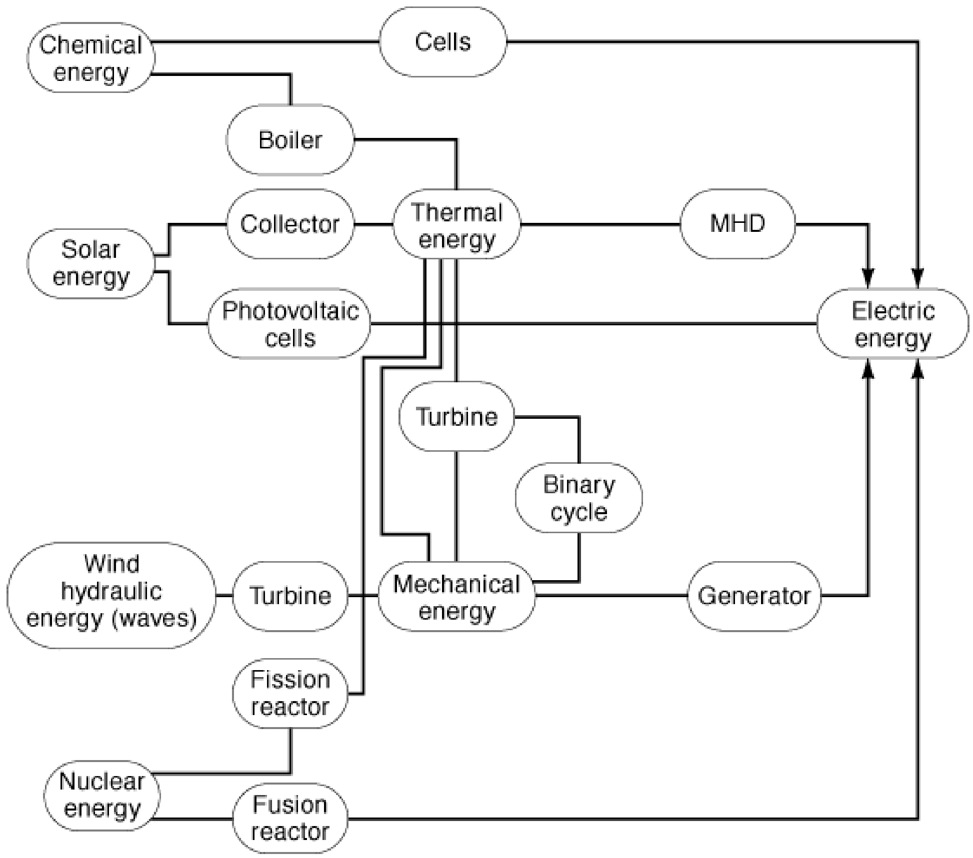

Fig. 1. Energy conversion paths.

compressor 3, storage tank 4, safety valves 5, manometers 6, purging valves 7, pressure line 8, valve 9, supply duct 10, safety valve 11, contact manometer 12, solenoid valve 13, pressure regulator 14, pneumatic valve 15, and cutoff valve 16.

Compressed air delivered by the compressor is stored in tanks. The diagram provides for successive supply of the two tanks. An automatic filling scheme for the two tanks is shown, made up of a manometer with contacts, giving the signal for solenoid valve to be opened, actuating in its turn a normally closed pneumatic valve through a pressure regulator. When a certain pressure is reached in the first tank, the contact manometer commands the opening of the supply to the second tank, which fills up from the first. When the pressure falls below a certain limit, the contact manometer will close off the supply, the pressure in the first tank will rise again, and the cycle will be repeated.

Figure 5 gives a diagram of the storage of compressed air generated by a wind compressor, where the compressed air is used to drive a gas turbine to produce electricity during maximum-load periods of a power system. For this purpose the traditional gas turbine type may be modified by kinematic separation of the compressor and turbine stages using gears (couplings), as shown in Fig. 6. In one of the separation stages the wind turbine actuates the compressor. The compressed air generated is stored in an accumulator, which may be a cavity or an exhausted natural-gas seam. In another stage, when electricity demands exceed the system

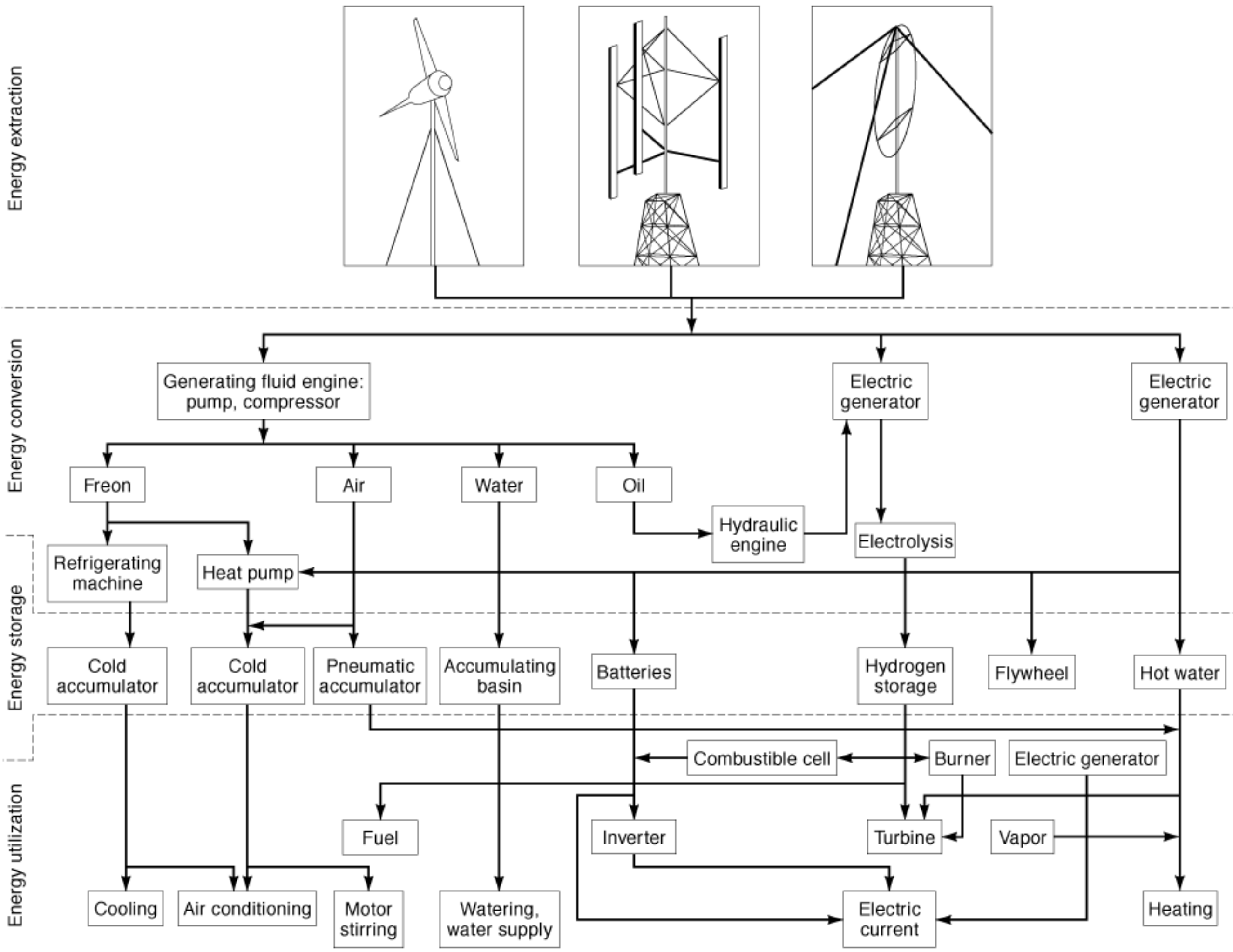

Fig. 2. Diagram of wind energy extraction, conversion, storage, and utilization.

capacity on basic load, the compressor is switched off and the generator is switched to the wind turbine. The burner is supplied with gas fuel and compressed air from the storage system.

On compression without heat loss (*adiabatic* compression), the air temperature increases. If compressed air is stored in a thermally well-insulated accumulator, we speak of adiabatic storage. When compressed air at this temperature is taken out and used to actuate a turbine, as in the previously mentioned case, less heat is required than if the storage were at ambient temperature. Therefore, adiabatic storage is more economical than isothermal.

Compressed Air Storage with Hydropneumatic Accumulators.

Description of the Accumulator. The hydropneumatic accumulator (Fig. 7) is made up of a pneumochamber (1) and a hydrochamber (2). Between the two chambers there is a 65 m to 70 m vertical level difference, corresponding to a 6.5 bar to 7.0 bar pressure, joined by a connecting incline (4). Both chambers are equipped with closing dams (5). At higher storage pressures, the level difference increases. The pneumochamber is

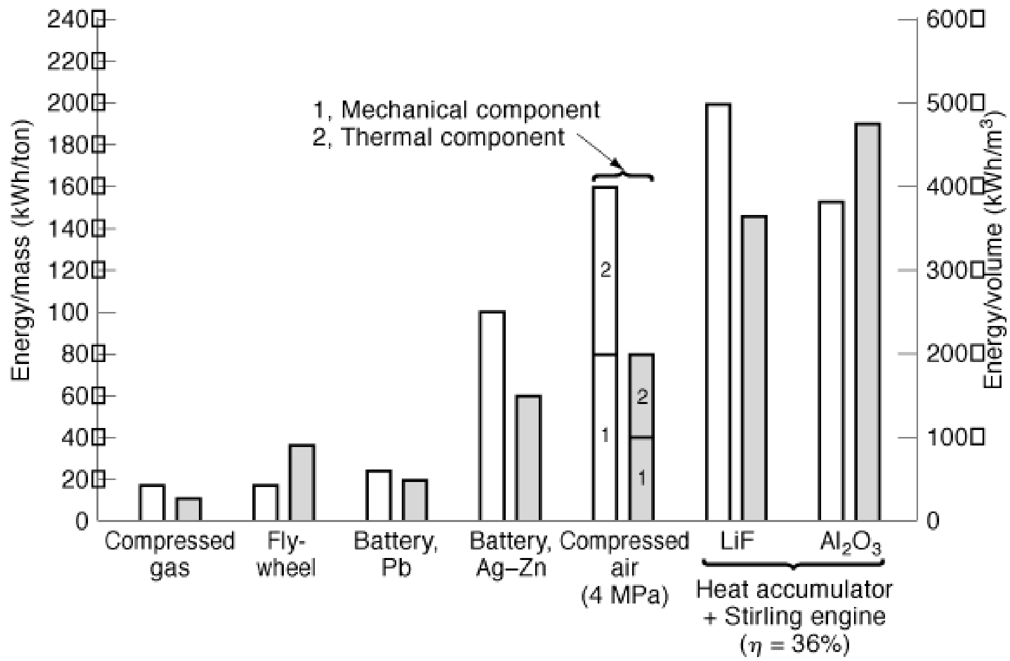

Fig. 3. Energy density for various energy accumulators.

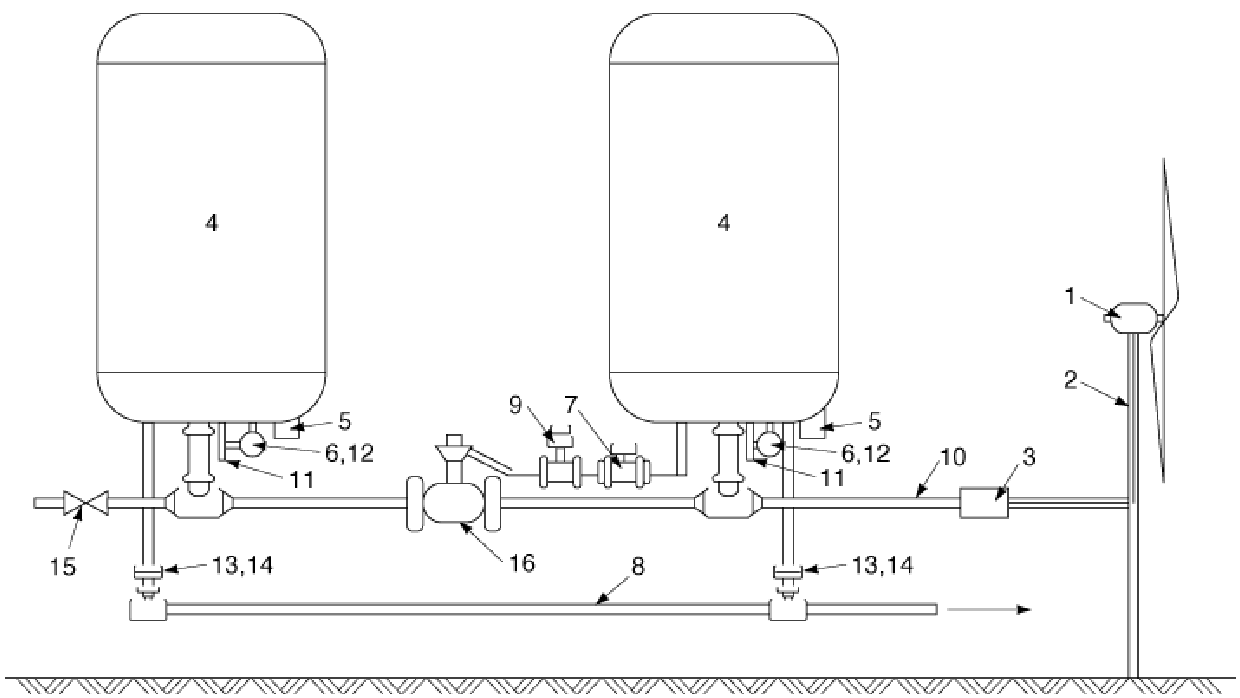

Fig. 4. Storage system for compressed air.

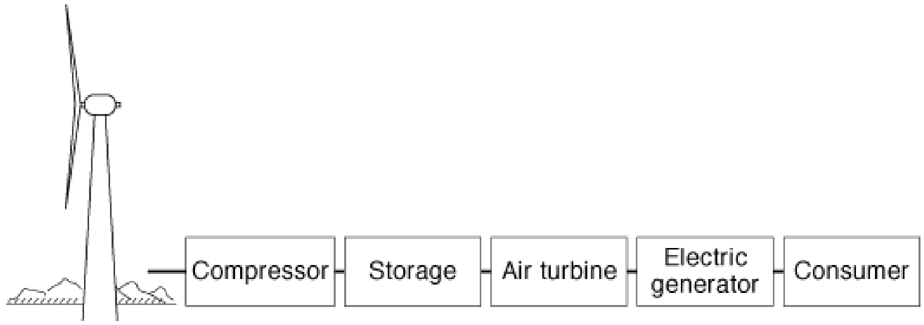

Fig. 5. Compressed air energy storage system.

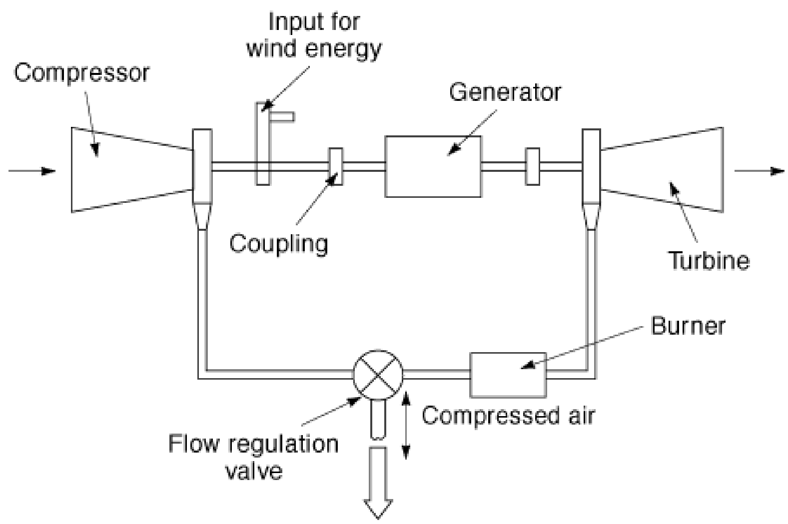

Fig. 6. Adiabatic storage of compressed air.

connected to the compressor station (6) by a compressed air supply duct (7), and to the main network by a distribution line (8). Both ducts are dimensioned to suit the average consumption. Water is supplied to the hydrochamber by a supply duct (9).

A hydraulic connection (11) exists between the pneumochamber and the hydrochamber, made up of metal ducts, with mining workings forming a hydraulic arrester in the lower part (12).

All ducts are fitted with shutoff valves (14), permitting the hydropneumatic accumulator to be isolated for inspection and maintenance as necessary. Between the compressed air supply duct and the distribution line there is a bypass (15), which may be used to short-circuit the hydropneumatic accumulator.

The pneumochamber must be carefully designed to avoid water or compressed air losses. Proper sealing is decisive for good operation and for the economic efficiency of the accumulator. With fissured rocks or with medium-hard and soft rocks, consolidation and sealing are effected by cement-milk or synthetic-resin injections.

The resistance dam closing the pneumochamber (1) must be specially constructed for durability and sealing, as it is crossed by the compressed air supply and distribution ducts, transducers for working and maximum levels, and an access opening where personnel can enter for control and maintenance. Optionally the resistance dam may be fitted with a water-gauge glass, which, during test periods, may provide data

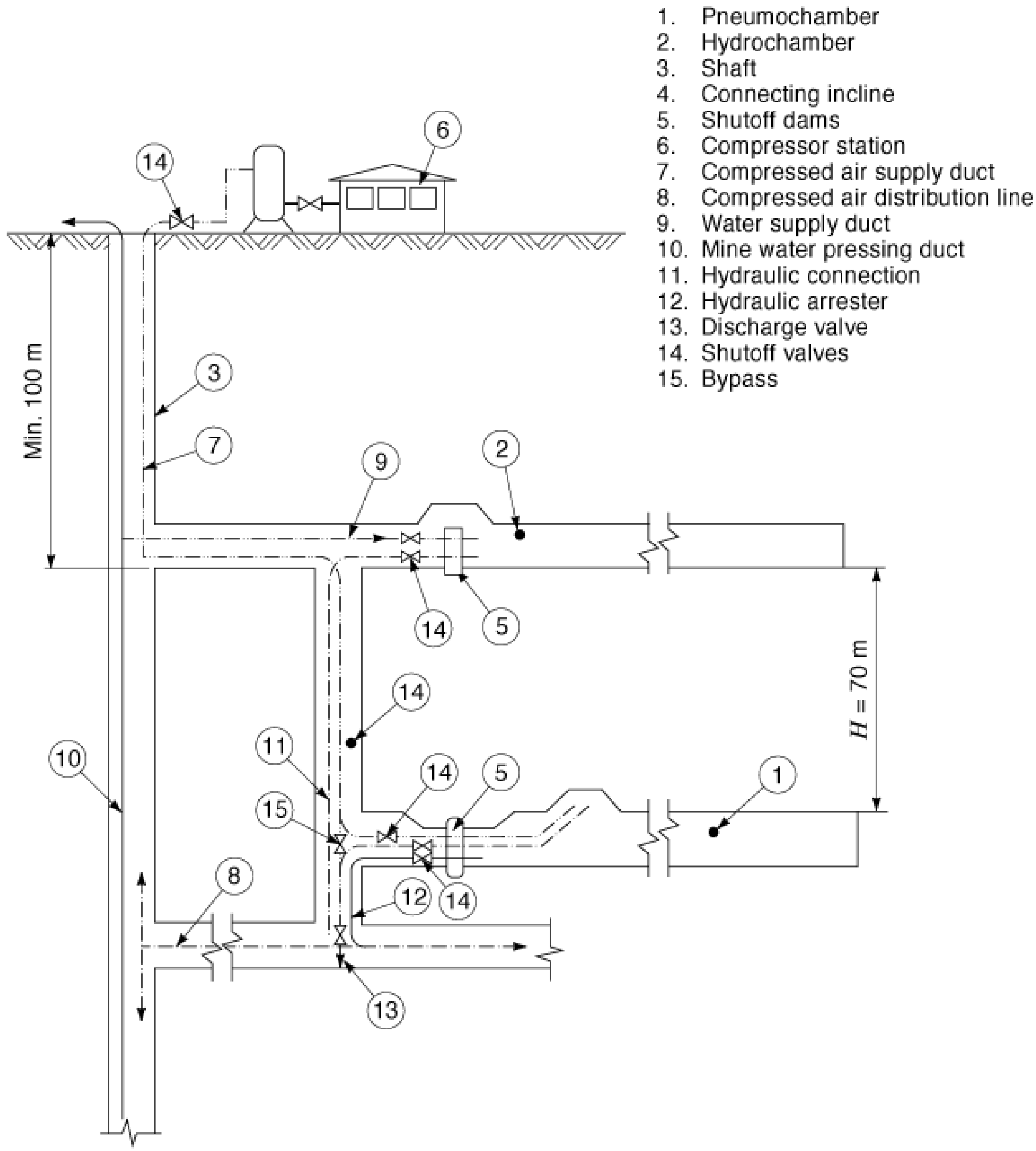

Fig. 7. The hydropneumatic accumulator.

about the adjustment or functioning of the signaling and monitoring equipment. The latter is essential for the protection of the accumulator, and its operation is programmed in synchrony with the compressor station working periods.

According to the general features presented, several types of accumulators may be devised, adapted to the local geological and mining conditions, to existing or planned mine workings, and to the compressor station's capacity.

In order to cut down investment costs, in building a hydropneumatic accumulator it is recommended to reuse decommissioned mining or geological research workings.

Operating Principle. A hydropneumatic accumulator is a complex of mining workings providing for the storage of a large volume of compressed air (1500 m^3 to 8000 m^3) at a constant pressure of 6.5 bar to 8.5 bar, which can be utilized by consumers in case the compressor station is out of service. Thus, idle running of compressors in low-consumption periods is avoided, and optimum parameters of compressed air in peak periods are ensured when the consumption required is higher than the capacity of the compressors in operation. Two distinct stages occur in the operation of the hydropneumatic accumulator:

- Accumulation of compressed air in the pneumochamber, in periods when the flow of the compressor station exceeds the flow absorbed by consumers, during which water is evacuated into the hydrochamber.
- Return of compressed air, while water from the hydrochamber flows into the pneumochamber and air consumption continues. In this stage the compressor station may run at a reduced flow rate or may be completely stopped.

To summarize, the hydropneumatic accumulator is a compressed air reservoir with variable volume and constant pressure, ensured by the potential energy of the water column joining hydrochamber and pneumochamber.

By the monitoring and signaling installation the current state of the pneumochamber is reported to the compressor station so as to stop or start the compressors. The moments at which the compressors are started and stopped being thus clearly determined, idling is completely eliminated, leading to a valuable reduction in energy consumption.

At the end of the first stage in the operation of the hydropneumatic accumulator, if the compressors are not stopped in time, there is a risk that the compressed air in the pneumochamber will pass into the hydrochamber through the hydraulic connection, where it can spray or push water out of the hydrochamber, causing unbalanced operation and possible severe damage to the surrounding mining workings. To prevent such accidents the accumulator is fitted with a hydraulic arrester made up of an additional 10 m to 13 m water column, for the evacuation of which the pressure in the accumulator is increased over the working pressure of the compressors by 1 bar (10 m H_2O).

Dimensioning Criteria for the Accumulator. The main operational parameters to be established for a hydropneumatic accumulator are the pneumochamber volume, the hydrochamber volume, the pressure in the pneumochamber, and the height between chambers.

The *pneumochamber volume* may be assessed according to several criteria:

- When the active capacity of the compressor station has been determined as a function of the maximum consumption of the users, we have

$$V = (Q_{\max} - Q_{\text{st}})\tau \tag{1}$$

where

V = pneumochamber volume (m^3)

$Q_{\max}$ = maximum compressed air consumption (m^3/min)

Q_{st} = actual flow of active compressors (m^3/min)

τ = time of maximum-consumption requirement (min)

- One may use the following empirical function of the active power of the compressor station: for each kilowatthour, 1.5 m^3 of accumulation buffer reservoir volume is needed at 7 bar pressure (1).

- The volume may be chosen according to the minimum capacity of operation of the compressor station determined by the minimum primary energy (wind) availability. In periods of minimum availability, only a few of the compressors may be operated or the entire station may cease to function. If C_{mf} is the minimum capacity for operation in minimum-availability hours, Q_{m} is the average consumption, τ is the longest duration of the primary energy deficit, and p is the working pressure of the accumulator, then the pneumochamber volume in cubic meters will be

$$V = \frac{(Q_{\mathrm{m}} - C_{\mathrm{mf}})\tau}{p} p_{\mathrm{N}} \tag{2}$$

The *hydrochamber volume* is an established function of the pneumochamber volume, being approximately 10% larger to compensate possible water losses and to accommodate water infiltration.

The *pneumochamber pressure* is function of the rated pressure of the compressor station and is determined with the relationship:

$$p_{\mathrm{p}} = p_{\mathrm{s}} - \Delta p_l - \Delta p_\varepsilon \tag{3}$$

where

p_{p} = pressure in pneumochamber (bar)
p_{s} = pressure in compressor station (bar)
Δp_{l} = sum of linear losses (bar)
Δp_ε = sum of local losses due to supports, bends, or cross-section alteration (bar).

The condition for the accumulator to be able to operate is that the compressed air pressure supplied by the compressor station at the entrance to the pneumochamber should be higher (by 0.2 bar to 0.3 bar) than the pressure within. If this condition is not met, hydraulic compensation is no longer possible and the accumulation volume is thus reduced to zero due to the water flooding in.

The *height between chambers* is an established function of the pressure in the pneumochamber, according to the relationship

$$H = \frac{p_{\mathrm{p}} - p_0}{\gamma} \tag{4}$$

where

H = height between the pneumochamber and the hydrochamber (m)
p_{p} = compressed air pressure in the pneumochamber (N/m^2)
p_0 = air pressure above water in the hydrochamber, (N/m^2)
γ = volumetric weight of water (N/m^3).

The height determined as above is a maximum, and it refers to the working height of water; the geometrical height is lower by the differences between minimum and maximum water levels in the chambers in the two working stages.

Besides the previously presented storage systems, two other systems applied in the case of natural gases are considered by us as particularly efficient. These are:

- Storing compressed air in the main network
- Storing compressed air in a high-pressure annular collector–distributor

Storing Energy by Compressed Air for Thermal Power Stations. To produce low-cost electric energy, thermal power stations must be operated continuously and, as far as possible, at optimum load with respect to energy (maximum efficiency, minimum specific consumption, and low pollution). The performance

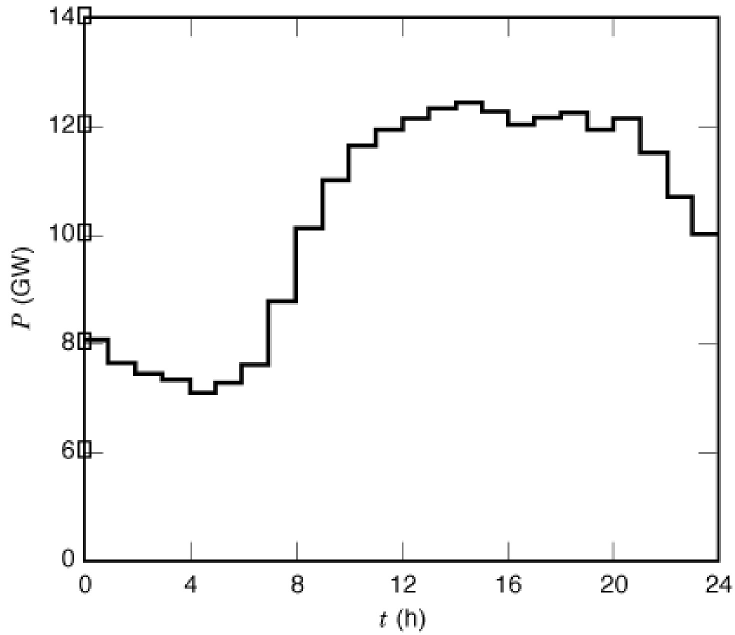

Fig. 8. Diagram of daily load (P=power; t=time).

of distribution networks is also favored by these conditions. Unfortunately, however, they are rarely met in practice. In many cases, most of the time, the electricity consumption in an area is almost twice as high in the daytime as at night. The diagrams presented in Figs. 8 and 9 show that there is an unbalance both in daily and in weekly loads.

In most cases, peak loads are covered by the use of gas turbines. Gas-turbine installation costs are low, but the fuels used in them (natural gas or petroleum) are expensive and liable to be depleted, with the further disadvantage of not being able to use the basic stations' nighttime power reserves.

For large storage capacities ($\geq$ 1500 MWh) and economical operation, hydraulic and pneumatic storage have proved to be the most advantageous. Hydraulic storage is recommended for areas with uneven topography. In flat regions pneumatic storage is better.

Thermoelectric power stations with air reservoirs have the following advantages:

- Improvement of daily and weekly load balance and a more rational use of existing stations
- The possibility of using secondary energy resources at the stations
- Fuel savings compared to other peak stations
- The possibility of coupling systems for the use of renewable energy resources to thermoelectric stations.

The most advantageous economic and functional conditions for compressed air storage reservoir arrangements are offered by underground saline rock formations. Air storage containers can also be excavated in hard rocks such as granite, gneiss, sandstone, or limestone. Since these are expensive operations, air reservoirs should be hydraulically compensated (water column) to minimize the volume necessary for the reservoir. In some cases, natural or existing artificial cavities (abandoned mine workings, emptied petroleum or natural-gas reservoirs) can be used as air reservoirs.

Various schemes for pneumatic energy storage at thermoelectric stations are now compared with respect to the storage cycle and choice of reservoir.

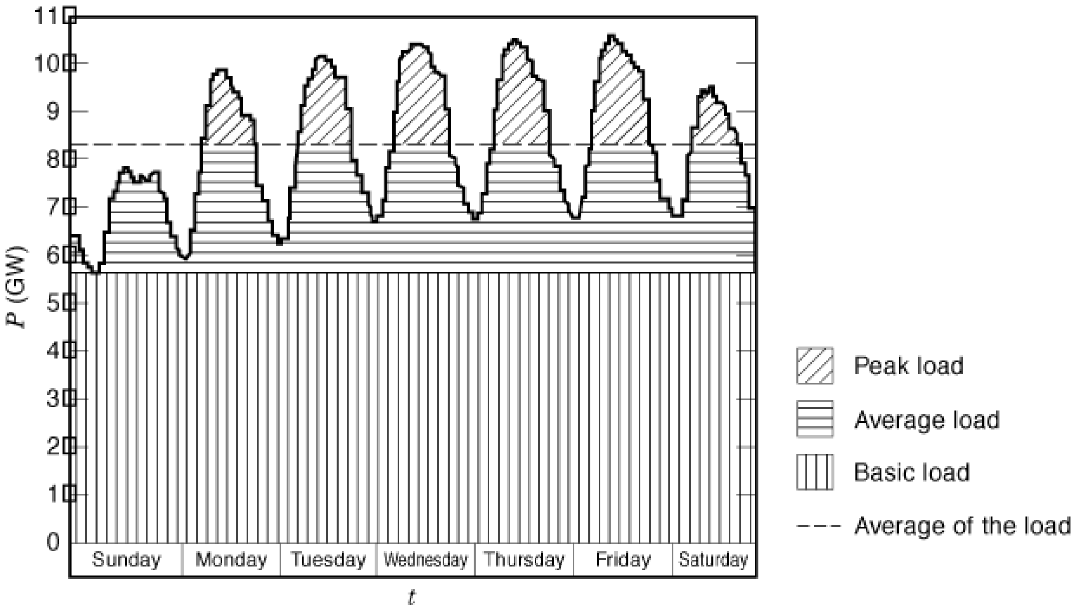

Fig. 9. Diagram of weekly load.

Adiabatic Storage Installations. An adiabatic installation (Fig. 10) is the pneumatic equivalent of a hydraulic pumping and turbocompressing station, representing the ideal case. No heat exchange with the exterior takes place (except in the cooler F, intended to removing humidity from the compressed air).

During the loading stage, compressed air without intermediary cooling yields heat to a heat accumulator before passing to the reservoir. At discharge and commissioning of the turbine, the accumulator warms up the air coming from the reservoir before its detention in the turbine. The input temperature in the turbine is determined by the final compressor temperature, which is lower ($t_{11} < t_4$, Fig. 10). The turbine input temperature is therefore a function of the reservoir pressure.

Nonadiabatic Storage Installation Fitted with a Combustion Chamber. In practical cases, pneumatic-storage stations use nonadiabatic storage installations (Fig. 11) with low-pressure and high-pressure combustion chambers. In the case of nonadiabatic installations, turbine input temperatures are independent of reservoir pressure, being determined by the fuel injected upstream of the turbine.

Comparison Criteria for Different Storage Cycles. Four characteristic values for comparing various thermodynamic cycles of air-resevoir stations are suggested in the literature (2):

$$\alpha = \frac{E_{\text{Gen}}}{m_{\text{a}}} = \frac{\text{alternator-supplied energy}}{\text{air mass passing through turbine}} \qquad (\text{kJ/kg})$$

$$\beta = \frac{E_{\text{Gen}}}{V_{\text{s}}} = \frac{\text{alternator-supplied energy}}{\text{reservoir volume}} \qquad (\text{kJ/m}^3)$$

$$\gamma = \frac{E_{\text{f}}}{E_{\text{Gen}}} = \frac{\text{energy of the fuel used}}{\text{alternator-supplied energy}} \qquad (\text{kJ/kJ})$$

$$\delta = \frac{E_{\text{Mot}}}{E_{\text{Gen}}} = \frac{\text{motor-absorbed energy}}{\text{alternator-supplied energy}} \qquad (\text{kJ/kJ})$$

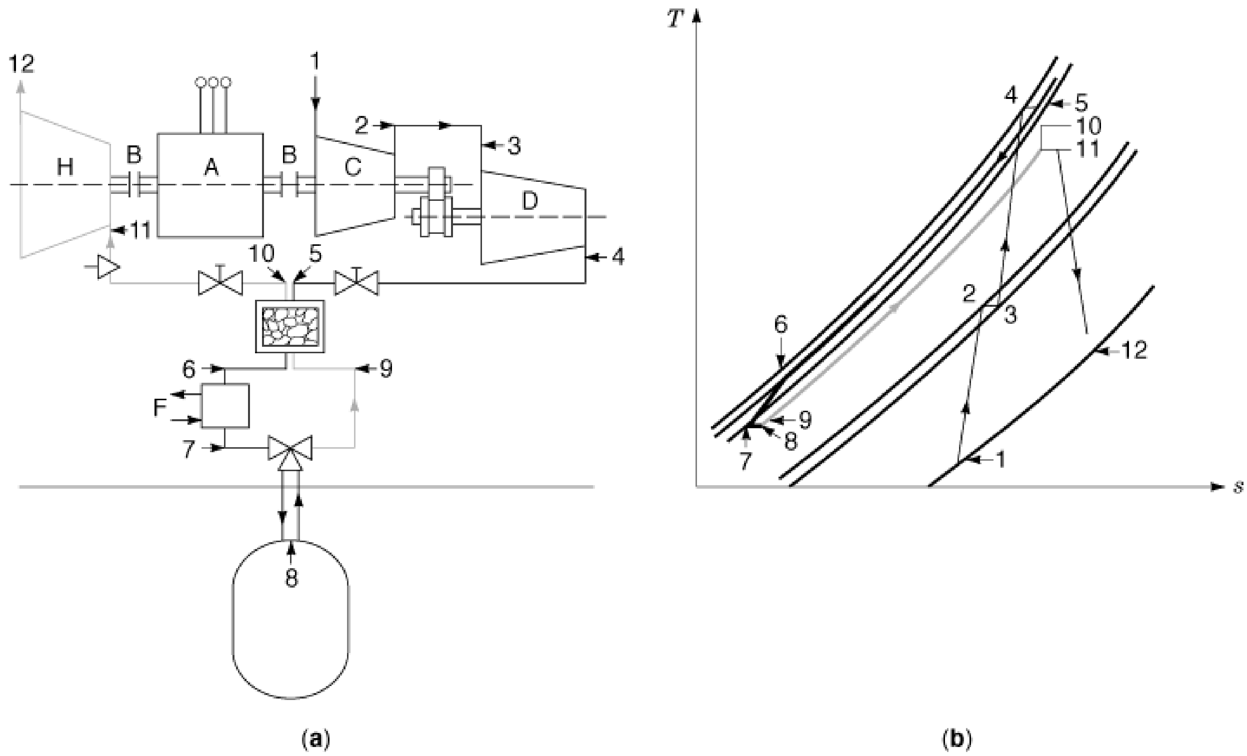

Fig. 10. Adiabatic storage installation. (a) Schematic diagram: A, alternator–motor; B, coupling; C, low-pressure compressor; D, high-pressure compressor; E, heat accumulator; F, cooler; G, air reservoir; H, turbine. (b) Entropy diagram: T=temperature; s=specific entropy.

The first two characteristic ratios (α and β) are related to construction expenses, and the last two (γ and δ) to operation expenses, of an air-reservoir station. The ratio α determines the turbine and compressor dimensions, and β the size of the air reservoir. The ratio γ determines the amount of fuel needed in the combustion chamber, and δ the necessary pumping energy.

To appreciate the cooperation between the air-reservoir station and the electric network or basic station, which supplies the energy necessary for the compressor drive, the *total efficiency* η_{RT} is relevant. It equals the total fuel energy (kJ) required in the basic station and in the air-reservoir station to obtaining 1 kJ at the alternator terminals. From η_{RT} the loss coefficient ε can be derived, expressing the total energy lost to the surroundings as heat. The expressions for the two indicators are

$$\eta_{\mathrm{RT}} = \frac{E_{\mathrm{Gen}}}{E_{\mathrm{f1}} + E_{\mathrm{f2}}} = \frac{1}{\dfrac{E_{\mathrm{Mot}}}{E_{\mathrm{Gen}}}\dfrac{1}{\eta_{\mathrm{B}}} + \dfrac{E_{\mathrm{f2}}}{E_{\mathrm{Gen}}}}$$

$$\varepsilon = \frac{E_{\mathrm{Amb}}}{E_{\mathrm{Gen}}} = \frac{1}{\eta_{\mathrm{RT}}} - 1$$

where

E_{Gen} = energy delivered at alternator terminals at the air-reservoir station

E_{Mot} = energy supplied to compressor motor at the air-reservoir station by the basic station

$\eta_{\mathrm{B}}=E_{\mathrm{Mot}}/E_{\mathrm{f_1}}$ = basic station efficiency versus motor terminals.

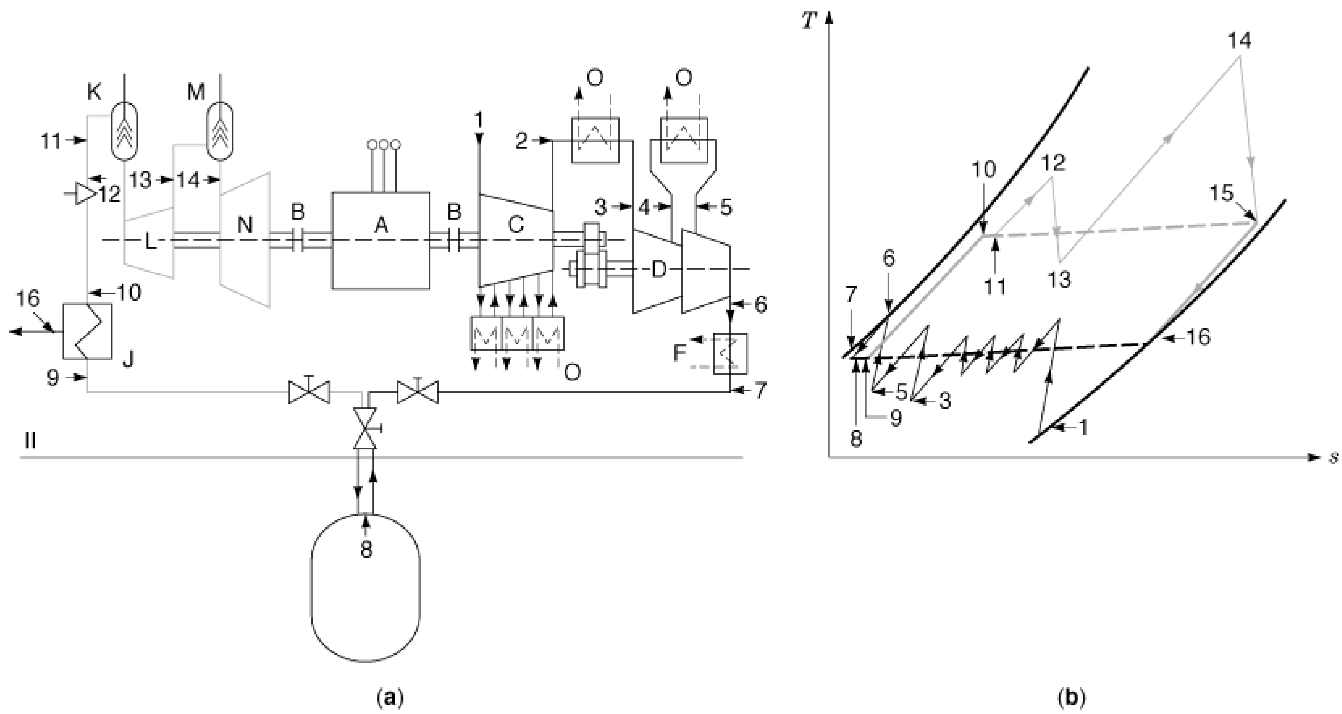

Fig. 11. Nonadiabatic storage installation fitted with low-pressure and high-pressure combustion chambers. (a) Schematic diagram (see also Fig. 10): J, recuperator; K, high-pressure combustion chamber; L, high-pressure turbine; M, low-pressure combustion chamber; N, low-pressure turbine; O, intermediary cooler. (b) Entropy diagram.

Table 1. Use Potential of Different Energy Forms

Energy Form	Exergy/Energy (%)
Potential energy	100
Kinetic energy	100
Electrical energy	100
Chemical energy	≈100
Nuclear energy	≈95
Sunlight (direct, diffuse)	73 to 92
Hot steam (200°C)	≈40
Space-heating water (90°C)	≈20
Residual warm water (20°C)	≈3

E_{f_1} = fuel energy for basic station
E_{f_2} = fuel energy for air-reservoir station
E_{Amb} = heat lost to the surroundings

The efficiencies of the adiabatic and nonadiabatic schemes are shown in Table 2.

A few remarks should be added on the financial aspects of these comparative values. The total costs associated with an air-reservoir station can be analyzed as follows (for ten years of turbine operation with reservoir load):

Machines. 40% to 50%

Table 2. Comparison of the Characteristic Ratios (air storage reservoir at constant pressure)

Quantity	Unit	Valve	
		Adiabatic	Nonadiabatic
Reservoir pressure	MPA	2 to 5	5
$\alpha = E_{Gen}/m_a$	kJ/kg	370 to 630	700 to 830
$\beta = E_{Gen}/V_s$	MJ/m^3	7.8 to 33	37 to 43.5
$\gamma = E_f/E_{Gen}$	kJ/kJ	0	1.19 to 1.15
$\delta = E_{Mot}/E_{Gen}$	kJ/kJ	1.35 to 1.30	0.75 to 0.65
η_{RT}	%	22 to 30	30 to 36
$\varepsilon = E_{Amb}/E_{Gen}$	kJ/kJ	2.4 to 3.5	1.7 to 2.3

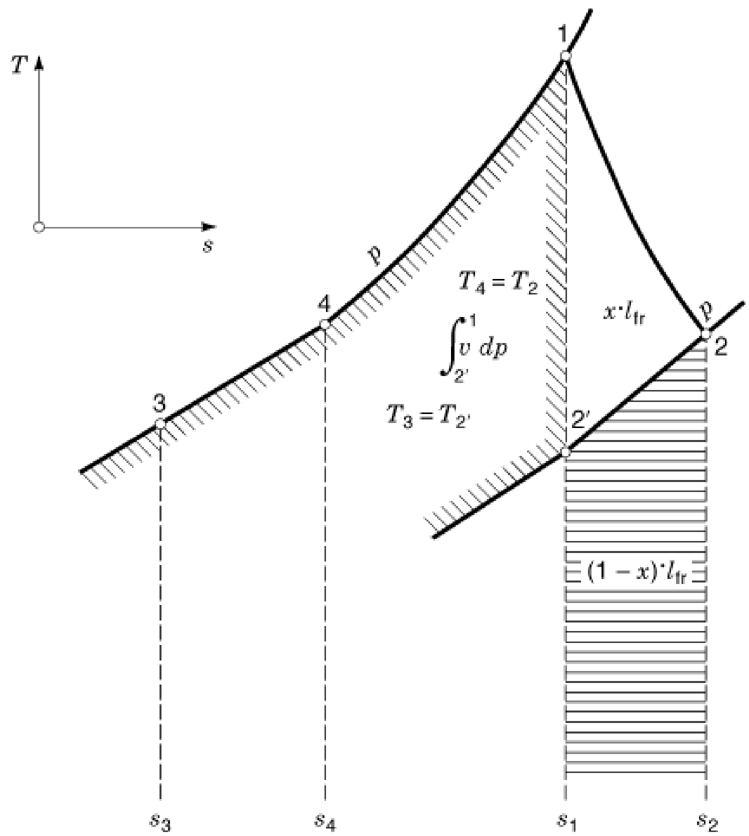

Fig. 12. Representation of compressed air expansion in a T–s diagram.

Air reservoir. 30% to 15%
Annexes. 30% to 35%

The low values for the reservoir refer to reservoirs in saline rocks. If several groups of machines were connected to the same reservoir, its share in the overall cost would be even smaller.

In conclusion one may say that the energy storage installations described in this section are economic.

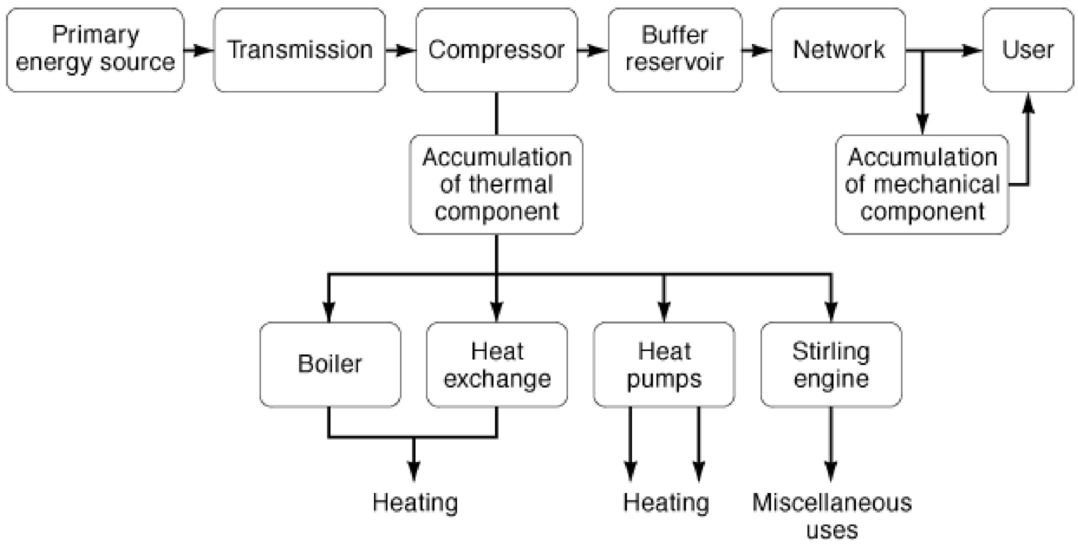

Fig. 13. Energy storage and delivery by compressed air.

Compressed Air—Energy Storage Medium and Energy Transfer Medium

Advantages and Disadvantages of Compressed Air for Use in Energy Transfer and Storage. Compared to other energy storage systems, compressed air shows the following advantages:

- It is an elastic, flexible, and comparatively safe medium.
- It has a reasonable capacity for energy per unit volume or weight, and it becomes competitive at high pressure when mechanical storage and thermal storage are combined.
- It uses part of the energy lost to irreversibility (*xl* in Fig. 12) as kinetic energy during its flow.
- By a comprehensive assessment it is less polluting than other energy storage systems.
- It is safe in explosive atmospheres.
- In underground mine workings, when exhausted from pneumatic equipment, it cleans the atmosphere, and in an emergency (collapses, people trapped underground) the pneumatic network allows trapped staff to be supplied with air, water, and food.
- It allows a large number of accumulation–discharge cycles without diminishing rated capacity.
- It offers simple construction and high operating reliability.
- It allows multiple use of pneumatic energy.

Low energy efficiency is its main disadvantage, both in industrial uses and in traditional storage systems. The losses within the process of compressed air generation, transport, storage, and use are as follows: 18% loss by friction in the compressor, 10% air losses due to compressor leakage, 18% internal energy losses by cooling in storage, and 34% losses in adiabatic expansion. These losses sum to 80%, so that 20% remains available.

Ignoring constructive improvements of the compressor (losses due to friction and leakage diminished) and focusing on recovering the thermal component in compressed air generation and on as complete as possible utilization of the adiabatic expansion (pneumatic expansion engines, refrigeration plants with air), we can state that the available energy may be increased by about 40%, so that the utilization coefficient for pneumatic energy should be competitive with that for electricity.

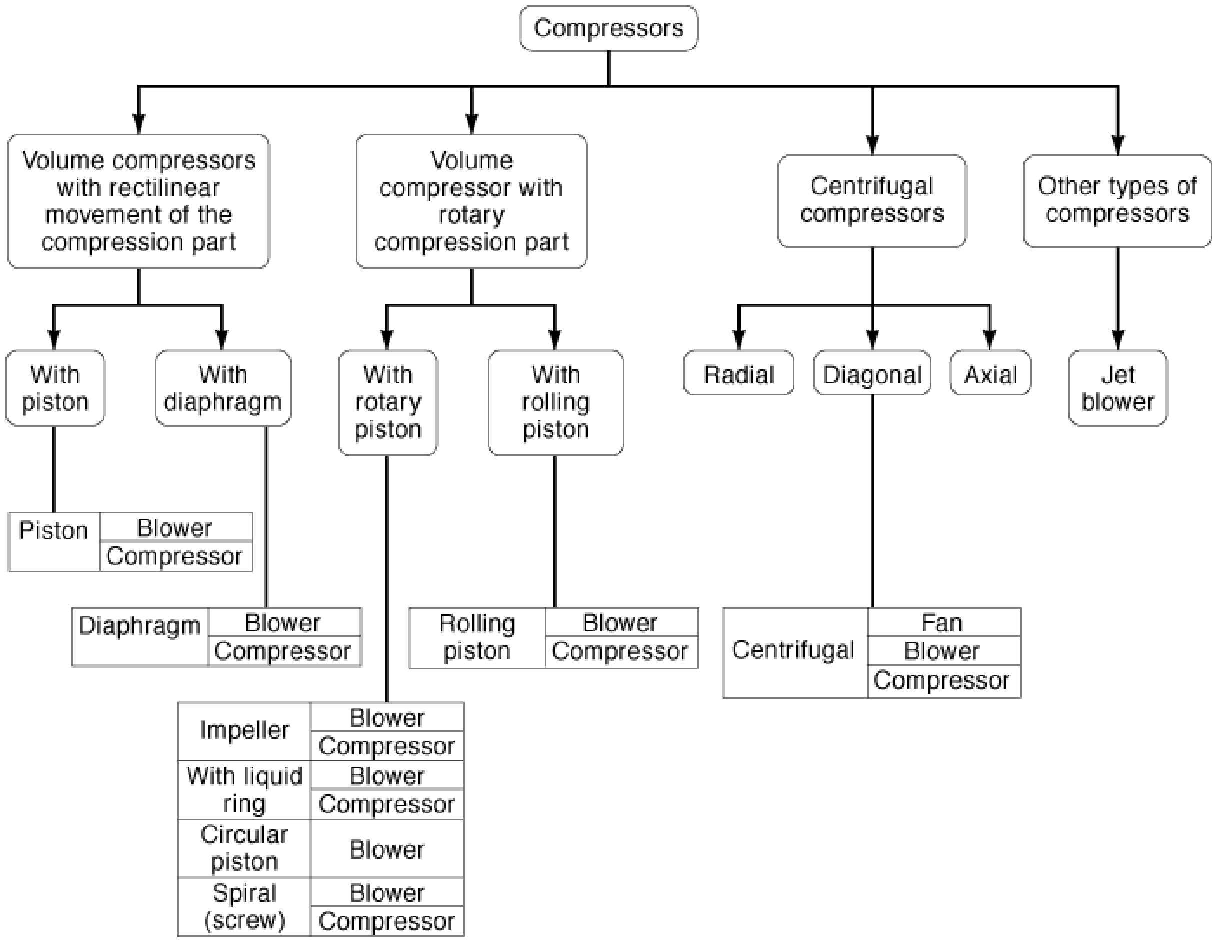

Fig. 14. Classification of compressors. In volume compressors the flows do not depend on the exhaust pressure; in centrifugal compressors the exhaust flow depends on the compression.

General Scheme for Compressed Air Energy Storage. The diagram in Fig. 13 represents a summary of the possible developments by which energy storage with compressed air could become economically attractive.

Types of Compressors and Fields of Use. The various types of compressors used to generate compressed air are presented in Fig. 14. In the selection of the compressor type, the conditions required by the primary energy source, the consumer, and the economic circumstances must be allowed for.

The limits of use for various types of compressors in terms of pressure and flow are presented in Fig. 15.

Exergetic Analysis of Processes from the Component Elements of Energy Storage.

General View. The notions of heat exergy and thermal agent exergy in continuous and stationary flow are natural consequences of the Gouy–Stodola law and are, as will be seen, useful in determining the losses caused by external irreversibility of thermal processes (3,4).

In a series of cases found in engineering (e.g., compressors and detentors with pistons), separate identification of the influence exerted by various physical state values on the exergy of a thermal agent is

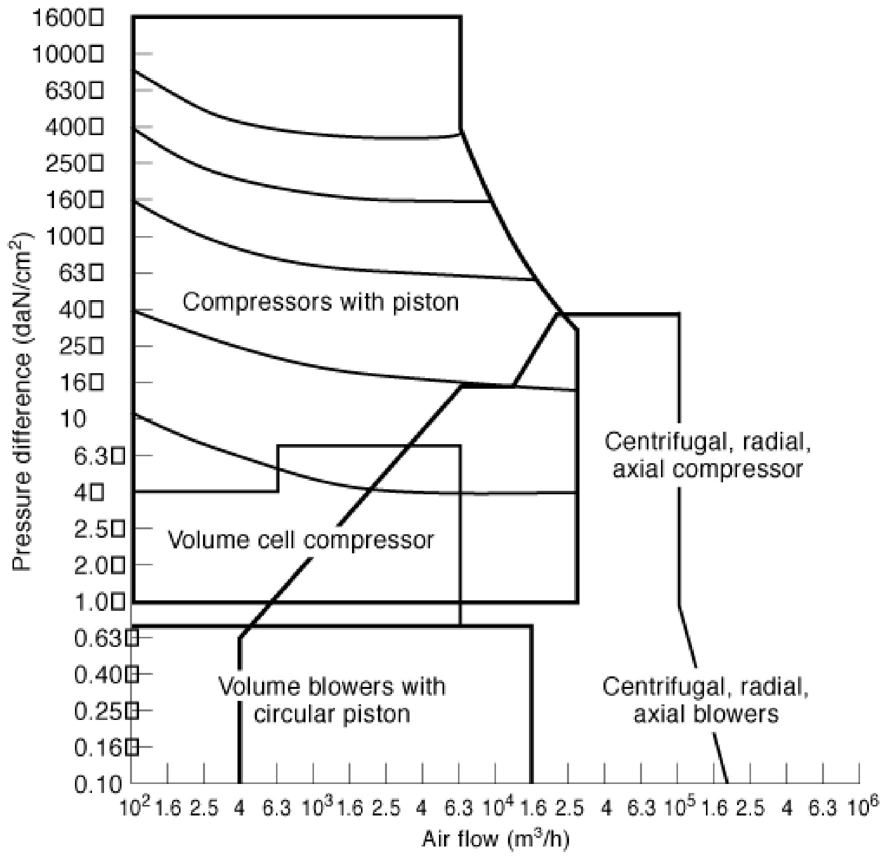

Fig. 15. Domains of use for various types of compressors in terms of pressure and flow.

necessary. To develop this idea, the case of the exergy e of a perfect gas, characterized by the parameters $p>p_0$ and $T>T_0$, is considered. Denoting the isobaric specific heat of the gas by c_p and the gas constant by R, we have

$$\begin{aligned} e &= i - i_0 - T_0 \cdot (s - s_0) = c_p(T - T_0) - T_0\left(c_p \ln \frac{T}{T_0} - R \ln \frac{p}{p_0}\right) \\ &= \left(c_p(T - T_0) - c_p T_0 \ln \frac{T}{T_0}\right) + RT_0 \ln \frac{p}{p_0} \end{aligned} \tag{5}$$

where

i = specific enthalpy
s = specific entropy and subscript 0 refers to the surroundings.

The exergy e is seen to consist of two components: (a) one that depends only on the temperature of the fluid, called the thermal component e_T, and (b) one that depends only on its pressure, called the mechanical

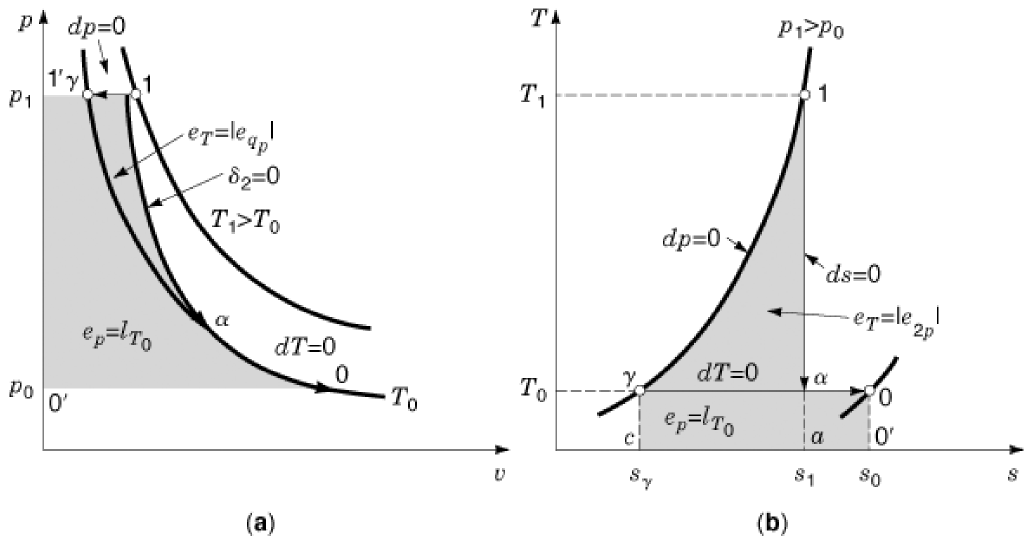

Fig. 16. Representation in p–v and T–s diagrams of the two exergy components.

component e_p (4). Consequently,

$$
\begin{aligned}
e_T &= c_p T_0 \left(\frac{T}{T_0} - 1 - \ln \frac{T}{T_0} \right) \\
e_p &= R T_0 \ln \frac{p}{p_0}
\end{aligned}
\tag{6}
$$

and thus

$$e = e_\mathrm{T} + e_\mathrm{p} \tag{7}$$

A thorough analysis of the structure of the two components leads to the following interpretations:

- The thermal component represents the absolute value of heat exergy exchanged by the gas with the exterior in an isobaric temperature change from T to T_0.
- The mechanical component represents the isothermal work exchanged by the gas with the exterior at temperature T_0 during a pressure change from p to p_0.

The p–v [Fig. 16(a)] and T–s [Fig. 16(b)] graphs clarify the significance of the two exergy components.

Examining the expressions for the components e_T and e_p of exergy, we see that the thermal fluid can be brought from state 1 to equilibrium with the environment in the following stages:

- Isobaric cooling 1–γ at pressure p_1 from temperature T_1 to T_0
- Isothermal expansion γ–0 at temperature T_0 from pressure p_1 to p_0.

Noting that [Fig. 16(a)]

$$e_p = l_{T_0} = RT_0 \ln \frac{p_1}{p_0} = \text{area}(1'\gamma 00')$$

and taking into consideration the graphical significance of the exergy e_1, we have

$$e_T = |e_{qp}| = c_p(T_1 - T_0) - c_p T_0 \ln \frac{T_1}{T_0} = \text{area}(1\gamma\alpha 1)$$

In the T–s graph [Fig. 16(b)] one may notice that

$$e_p = \text{area}(c\gamma 00')$$
$$e_T = \text{area}(1\gamma\alpha 1)$$

Considering that

$$c_p = \frac{k}{k-1} R$$

where k is the adiabatic exponent of the gas, and representing graphically the dimensionless values

$$\bar{e}_p = \frac{e_p}{RT_0} = \ln \frac{p}{p_0}, \qquad \bar{e}_T = \frac{e_T}{RT_0} = \frac{k}{k-1}\left(\frac{T}{T_0} - 1 - \ln \frac{T}{T_0}\right) \tag{8}$$

the curves shown in Fig. 17 are obtained. It is to be noticed that unlike the component e_T, which is always positive [Fig. 17(a)] irrespective of the temperature range ($T>T_0$ or $T<T_0$), the component e_p is positive in the range of pressures $p>p_0$ and negative in the range $p<p_0$ [Fig. 17(b)].

Exergetic Method of Analysis of Processes within Pneumatic Networks. Exergetic analysis that includes the first and second laws of thermodynamics allows quantitative definition of the qualitative term *energy degradation*. Consequences of various thermodynamically irreversible phenomena can thus be calculated, leading to correct understanding of thermodynamic losses within a system. Exergetic analysis contributes to the definition of a *thermodynamic efficiency*, expressing the degree of perfection or thermodynamic quality of a system. The concept of exergy stands at the basis of this type of analysis: a nonconserved quantity with the dimensions of energy but with unlimited transformation capacity. In the case where the system is made up of a flowing fluid, the exergy of the system is equal to the difference between the initial value of the Gibbs function (free enthalpy) of the system and the value of the Gibbs function of the system when it cannot generate useful work by interaction with thermal reservoir, that is, when the system is in thermal and mechanical equilibrium with the reservoir (3).

In the particular case of compressed air flow through the sections of a pneumatic network, the main causes of internal and external irreversibility are:

(1) For internal irreversibility:

- Nonstatic character of expansions
- Internal friction between various parts of the fluid and between fluid and duct walls; friction and shocks due to eddies;

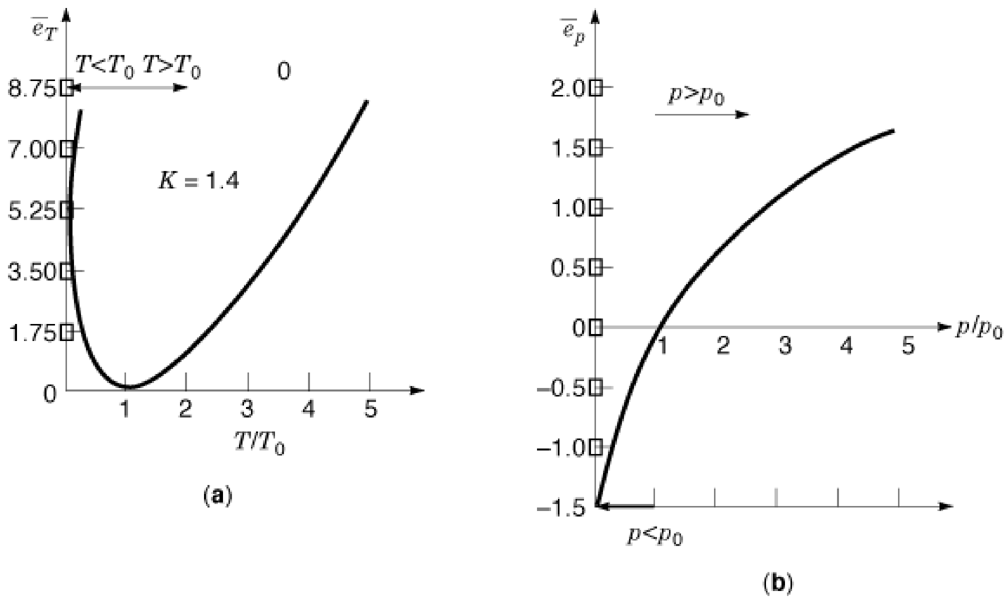

Fig. 17. Graphical representation of the dimensionless quantities _e_T and _e_p.

- Moisture condensing and partially obstructing the duct
- Mixing and homogenization of humid air, water, oil, dust, and rust
- Finite pressure and temperature differences generated between various parts of the fluid

(2) For external irreversibility:

- Heat exchange at finite temperature differences between fluid and environment
- Material transfer at finite pressure differences from fluid to environment due to flow losses caused by leakage.

Let $M_{\rm it}$ be the compressed air mass flow into a section; $M_{\rm et}$ the mass flow out of the section; $M_{\rm mt}$ the average mass flow through the section; $\Delta M = M_{\rm et} - M_{\rm it}$ the leakage; $T_{\rm ac}$ the average compressed air temperature; $p_{\rm m}$ the average compressed air pressure; i_0 and s_0 the specific enthalpy and specific entropy of the environment, respectively; $i_{\rm m}$ and $s_{\rm m}$ the average specific enthalpy and specific entropy of compressed air; $l_{\rm fr}$ the specific frictional work; and $q_{\rm ext}$ the heat transfer to the environment. Then we have for the equation of exergy balance of a duct section

$$M_{\rm it}e_{\rm it} = M_{\rm et}e_{\rm et} - \Delta M e_{\rm m} - M_{\rm mt}q_{\rm ext}\left(1 - \frac{T_0}{T_{\rm ac}}\right) + M_{\rm mt}l_{\rm fr}\frac{T_0}{T_{\rm ac}} \qquad (9)$$

Inserting the exergy expression Eq. (5) for a flowing fluid in Eq. (9), we have

$$M_{\mathrm{it}}[(i_1 - T_0 s_1) - (i_0 - T_0 s_0)] = M_{\mathrm{et}}[(i_2 - T_0 s_2) - (i_0 - T_0 s_0)] \\ - \Delta M[(i_{\mathrm{m}} - T_0 s_{\mathrm{m}}) - (i_0 - T_0 s_0)] - M_{\mathrm{mt}} q_{\mathrm{ext}}\left(1 - \frac{T_0}{T_{\mathrm{ac}}}\right) + M_{\mathrm{mc}} l_{\mathrm{fr}} \frac{T_0}{T_{\mathrm{ac}}} \tag{10}$$

Let us move the term representing the exergy of the fluid coming out of the section to the left side of Eq. (10) and suppose that losses due to leakage take place through adiabatic laminar flow, kinetic energy variations being negligible, so that $i_{\mathrm{m}}=i_0$. Then the exergy losses for compressed air flow through the considered duct section will be given by the equation

$$M_{\mathrm{it}} e_{\mathrm{it}} - M_{\mathrm{et}} e_{\mathrm{et}} = \Delta M T_0 (s_{\mathrm{m}} - s_0) + M_{\mathrm{mt}} q_{\mathrm{ext}}\left(\frac{T_0}{T_{\mathrm{ac}}} - 1\right) + M_{\mathrm{mt}} l_{\mathrm{fr}} \frac{T_0}{T'_{\mathrm{ac}}} \tag{11}$$

From Eq. (11), substituting $s_{\mathrm{m}}-s_0=c_p \ln(T_{\mathrm{ac}}/T_0)-R\ln(p_{\mathrm{m}}/p_0)$, replacing l_{fr} with $(1-x)l_{\mathrm{fr}}$ from Fig. 8, noting that $T'_{\mathrm{ac}}=(T_2+T'_2)/2$, and recognizing the mechanical component of exergy dissipation by friction, we obtain the following relationship for the exergy losses along the analyzed section:

$$\Delta E = E_{\mathrm{it}} - E_{\mathrm{et}} = \Delta M c_p T_0 \ln\left[\frac{T_{\mathrm{ac}}}{T_0}\left(\frac{p_0}{p_{\mathrm{m}}}\right)^{(k-1)/k}\right] + M_{\mathrm{mt}} q_{\mathrm{ext}}\left(\frac{T_0}{T_{\mathrm{ac}}} - 1\right) \\ + M_{\mathrm{mt}} T_0 c_p \ln\frac{T_2}{T_1} + M_{\mathrm{mt}} T_0 R \frac{\Delta p}{p_1} + M_{\mathrm{mt}} T_0 R \frac{\Delta p_{\mathrm{obt}}}{p_1} \tag{12}$$

From the terms giving the exergy loss due to irreversible transfer thermal between the system and the thermal source,

$$\Delta E_{\mathrm{q}} = M_{\mathrm{mt}} q_{\mathrm{ext}}\left(\frac{T_0}{T_{\mathrm{ac}}} - 1\right) = \Omega_1 K (T_0 - T_{\mathrm{ac}})\left(\frac{T_0}{T_{\mathrm{ac}}} - 1\right)$$

one may notice that if all values entering the expression are assumed constant, then the exergy loss ΔE_{q} is the same irrespective of the evolution of the system to $T_{\mathrm{ac}}>T_0$ or $T_{\mathrm{ac}}<T_0$—an explicable fact, since the return of the system to thermal equilibrium with the thermal source, from temperatures equally spaced above and below the temperature of the source, implies the same loss of exergy. The terms on the right side of Eq. (12) represent exergy losses due to irreversible phenomena accompanying compressed air flow along the sections of the pneumatic mining network. The analysis of Eq. (12) may lead to a series of technical and organizational measures to reduce exergy losses to a technical and economic minimum.

The exergy losses due to leakage in pneumatic networks,

$$\Delta E_{\mathrm{M}} = \Delta M c_p T_0 \ln\left[\frac{T_{\mathrm{ac}}}{T_0}\left(\frac{p_0}{p_{\mathrm{m}}}\right)^{(k-1)/k}\right]$$

represent the largest share (65% to 75%) of the total losses. To reduce them, the following measures are advisable:

- Continual monitoring and elimination of leaks
- Providing a compressed air temperature (T_{ac}) along the section as close as possible to the ambient temperature (T_0), on condition that $T_{ac}>T_0$
- Providing a compressed air pressure along the section as close as possible to the ambient pressure, consistent with the pressures required by the consumers, as well as with the upper limit on ΔE_p.

The exergy losses induced by the irreversibility of thermal transfer between compressed air and environment are given by

$$\Delta E_{\mathrm{q}} = M_{\mathrm{mt}} q_{\mathrm{ext}} \left(\frac{T_0}{T_{\mathrm{ac}}} - 1 \right) = \Omega_1 K (T_0 - T_{\mathrm{ac}}) \left(\frac{T_0}{T_{\mathrm{ac}}} - 1 \right)$$

They may be reduced by the following measures:

- Reducing the lateral area of the section (Ω_l) (also useful in lowering investment costs and reducing ΔE_M, but leading to an the increase in ΔE_p, so that a compromise is needed regarding the optimum diameter)
- Reducing the global thermal transfer coefficient K
- Providing a compressed air temperature along the section as close as possible to ambient temperature.

The energy losses due to flows with friction,

$$\Delta E_{\mathrm{p}} = M_{\mathrm{mt}} T_0 \left(c_p \ln \frac{T_2}{T_1} - R \ln \frac{p_2}{p_1} \right)$$

may be reduced by:

- Providing an as high as possible pressure for the compressed air along the section
- Reducing pressure losses due to friction (Δp) by increasing the duct diameter
- Reducing pressure losses resulting from partial obstruction of the duct (Δp_{obt}) by removing humidity from the compressed air and cleaning the insides of the ducts when they are mounted
- Conveying compressed air along the section at the highest possible temperature, thus cutting down exergy losses corresponding to work done against friction. (not advisable for pneumatic mining networks, as it leads to considerable increase of other loss categories)

The exergetic analysis is completed with a synthetic indicator suitable for characterizing the quality of a thermal installation in the sense of thermodynamic energetics—exergetic efficiency. Exergetic efficiency is a measure of the degree of irreversibility of the transformations taking place in a thermal installation, being quantitatively expressed as the ratio of useful work (useful exergetic effect) to the exergy difference between entrance and exit of the installation (available exergy):

$$\eta_{\mathrm{ex}} = \frac{l_{\mathrm{util}}}{e_{\mathrm{i}} - e_{\mathrm{e}}} \tag{13}$$

In the particular case of pneumatic mining network sections, since compressed air does not generate useful work while in them, exergetic efficiency is expressed by the relationship

$$\eta_{\mathrm{ex}} = \frac{e_{\mathrm{et}}}{e_{\mathrm{it}}} = 1 - \frac{\Delta e}{e_{\mathrm{it}}} \tag{14}$$

Taking into account the relationship in Eq. (12), the expression for the exergetic efficiency for a pneumatic network section becomes

$$\eta_{\mathrm{ex}} = 1 - \frac{\Delta M c_p T_0 \ln\left[\frac{T_{\mathrm{ac}}}{T_0}\left(\frac{p_0}{p_{\mathrm{m}}}\right)^{(k-1)/k}\right] + M_{\mathrm{mt}} q_{\mathrm{ext}}\left(\frac{T_0}{T_{\mathrm{ac}}} - 1\right) + M_{\mathrm{mt}} T_0 c_p \ln\frac{T_2}{T_1} + M_{\mathrm{mt}} T_0 R \frac{\Delta p + \Delta p_{\mathrm{obt}}}{p_1}}{M_{\mathrm{it}}[(i_1 - i_0) - T_0(s - s_0)]} \tag{15}$$

Exergetic efficiency represents the effectiveness of a thermal process. As it refers to the thermodynamic potential corresponding to the state of the environment, it constitutes measure of actual process imperfections.

Study of Storage-Tank Filling Process. For an exergetic approach to the filling and emptying of storage tanks we use a method perfected by Bejan (5) and Radcenko (4). We model the storage tank as a cylinder with a piston and a valve. The diagram in Figure 18 shows the complex process of filling the cylinder. When the intake valve opens, a residual air quantity m_{e} is in the cylinder, taking up a dead volume V_{c} at parameters p_{c} and T_{c} [Fig. 18(a)]. The mass of air entering the cylinder during admission, $m_{\mathrm{ad}}=m_{\mathrm{a}}-m_{\mathrm{c}}$, occupies the volume V_1 in the intake duct at parameters p_1 and T_1; it is separated from the rest of the air by an imaginary piston under pressure p_1. At the end of the filling stage at V=const, the cylinder contains the mass of air $m_{\mathrm{a}}>m_{\mathrm{c}}$ at parameters $p_{\mathrm{a}}=p_1-\Delta p_{\mathrm{a}}$ and $T_{\mathrm{a}}>T_{\mathrm{c}}$ [Fig. 18(b)]. When the intake valve is closed, the mass of air in the cylinder becomes m_{a}, occupying volume V_{a} at parameters p_{a} and T_{a} [Fig. 18(c)].

Let e_{qa} be the exergy exchanged by the air in the filling process, e_{a} the exergy when filling is over, e_1 the exergy when admission starts, l_{ta} the mechanical work of the isochoric filling stage, and π_{ira} the exergy loss due to the internal irreversibility of the filling process. Then the following exergetic balance equation of the filling process is obtained:

$$e_{\mathrm{qa}} = e_{\mathrm{a}} - e_1 + l_{\mathrm{ta}} + \pi_{\mathrm{ira}} \tag{16}$$

where

$$e_{\mathrm{a}} - e_1 = i_{\mathrm{a}} - i_1 - T_0(s_{\mathrm{a}} - s_1) = \left[c_p(T_{\mathrm{a}} - T_1) - c_p T_0 \ln\frac{T_{\mathrm{a}}}{T_1}\right] - RT_0 \ln\frac{p_1}{p_{\mathrm{a}}} \tag{17}$$

Denoting

$$e_{\mathrm{q1a}} = c_p(T_{\mathrm{a}} - T_1) - c_p T_0 \ln\frac{T_{\mathrm{a}}}{T_1} \tag{18}$$

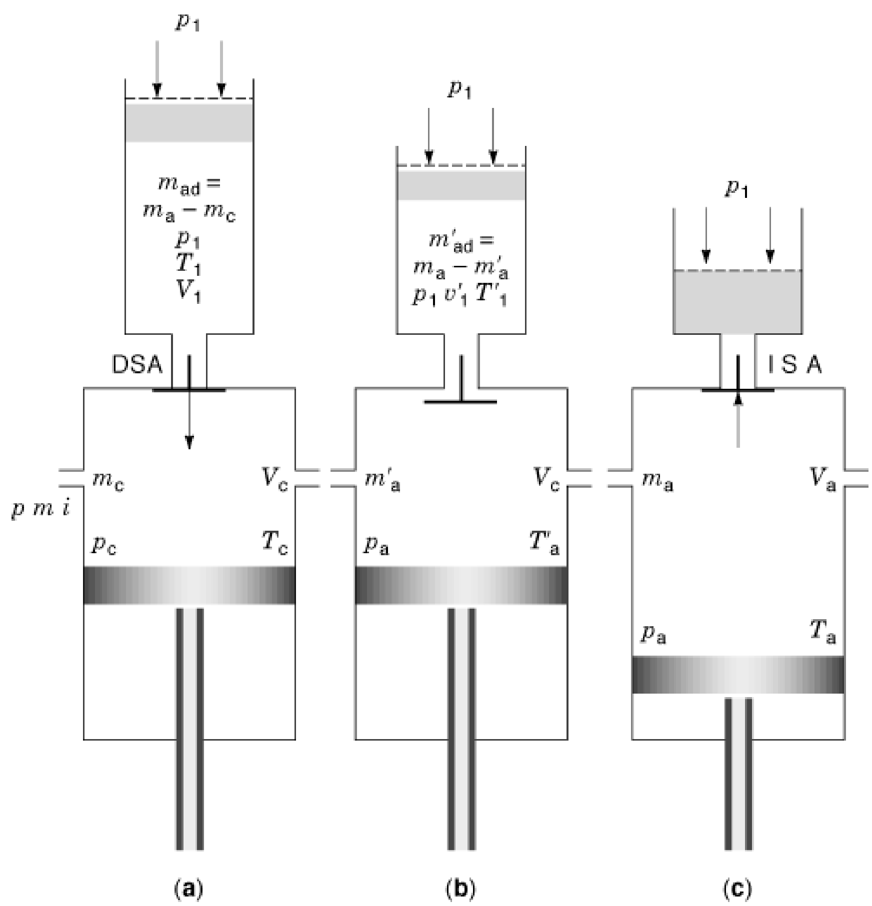

Fig. 18. Description of the tank filling process.

it is seen that

$$\pi_{1a} = RT_0 \ln = \frac{p_1}{p_a} = RT_0 \ln \frac{p_1}{p_1 - \Delta p_a} RT_0 \ln \frac{1}{1 - \Psi_a} \tag{19}$$

is the exergy loss due to laminar air flow in the cross section of the intake valve. Therefore

$$\pi_{ira} - e_{qa} = \pi_{1a} + |l_{ta}| - e_{q1a} \tag{20}$$

Denoting by

$$\pi_a = |l_{ta}| - e_{q1a} \tag{21}$$

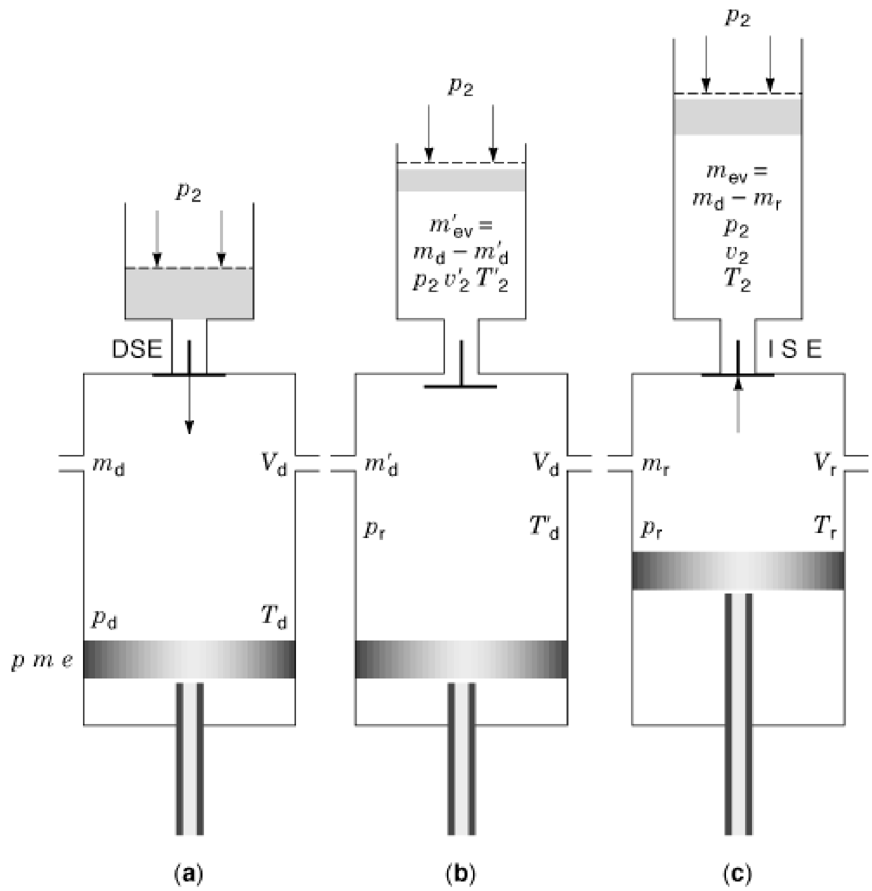

Fig. 19. Description of the tank evacuation process.

the exergy losses induced only by finite pressure and temperature differences during the filling process, we conclude that

$$\pi_{\text{ira}} - e_{\text{qa}} = \pi_{\text{a}} + \pi_{1\text{a}} \tag{22}$$

Study of Storage-Tank Evacuation Process. Figure 19 shows a diagram of the evacuation process. When the release valve opens, the mass of gas m_{d} in the cylinder occupies the total volume V_{d} at parameters p_{d} and T_{d} [Fig. 19(a)]. When evacuation is over, V=const, and the air mass $m_{\text{d}}-m'_{\text{d}}$ leaving the cylinder occupies a volume V'_2 at parameters p_2, T_2 in the exhaust duct, being separated from the rest of the exhausted air by an imaginary piston under a counterpressure p_2 [Fig. 19(b)]. When the exhaust valve is closed, the residual air mass m_{r} in the cylinder occupies the volume V_{r} at parameters p_{r} and T_{r}. The evacuated air mass $m_{\text{ev}}=m_{\text{d}}-m_{\text{r}}$, equal to the admitted mass m_{ad}, occupies a volume V_2 in the exhaust duct at parameters p_2 and T_2 [Fig. 19(c)].

Let e_{qe} be the exergy exchanged by the air in the evacuation process, e_2 the exergy when evacuation is over, e_{d} the exergy when evacuation starts, l_{t2} the mechanical work of evacuation, l_{Tod} the isothermal work of expansion, and π_{ire} the exergy loss due to internal irreversibility of evacuation. Then the exergetic balance equation for the evacuation process is

$$e_{\mathrm{qe}} = e_2 - e_{\mathrm{d}} + l_{\mathrm{td}} + \pi_{\mathrm{ire}} \tag{23}$$

where

$$e_2 - e_{\mathrm{d}} = i_2 - i_{\mathrm{d}} - T_0(s_2 - s_{\mathrm{d}}) = \left(c_p(T_2 - T_{\mathrm{d}}) - T_0 c_p \ln \frac{T_2}{T_{\mathrm{d}}}\right) - RT_0 \ln \frac{p_{\mathrm{d}}}{p_2} \tag{24}$$

We denote

$$e_{\mathrm{q}0} = c_p(T_{\mathrm{d}} - T_2) - c_p T_0 \ln \frac{T_{\mathrm{d}}}{T_2} \tag{25}$$

$$l_{\mathrm{T_{0d}}} = RT_0 \ln \frac{p_{\mathrm{d}}}{p_2} = RT_0 \ln \frac{p_{\mathrm{d}}}{p'} = RT_0 \ln \frac{p_{\mathrm{r}}}{p_2} \tag{26}$$

In the last relationship, the second term gives the exergy loss due to laminar gas flow in the section of the release valve:

$$\pi_{\mathrm{lc}} = RT_0 \ln \frac{p_{\mathrm{r}}}{p_2} = RT_0 \ln \frac{p_{\mathrm{r}}}{p_{\mathrm{r}} - \Delta p_{\mathrm{e}}} + RT_0 \ln \frac{1}{1 - \Psi_{\mathrm{e}}} \tag{27}$$

Consequently, observing that $e_{\mathrm{q0e}} < 0$ and denoting

$$l_{\mathrm{T_0d}} = RT_0 \ln \frac{p_{\mathrm{d}}}{p'} \tag{28}$$

it results that

$$\pi_{\mathrm{ire}} - e_{\mathrm{qe}} = l_{\mathrm{T_{0d}}} - l_{\mathrm{td}} - |e_{\mathrm{q}0}| + \pi_{\mathrm{le}} \tag{29}$$

Denoting by

$$\pi_{\mathrm{e}} = l_{\mathrm{T_0d}} - l_{\mathrm{td}}) - \left|e_{\mathrm{q}_0^{\mathrm{n}}}\right| \tag{30}$$

the exergy losses induced by finite pressure and temperature differences characterizing the evacuation process, we have

$$\pi_{\mathrm{ire}} - e_{\mathrm{qe}} = \pi_{\mathrm{e}} + \pi_{\mathrm{le}} \tag{31}$$

Combining with the above balances the exergetic balances of other components in the energy conversion and storage scheme, a thermal–economic calculation of the performance of the energy-conversion and -storage installation may be made.

Determination of the Accumulation Capacity of the Transport Section in Pneumatic Networks. An efficient energy storage system using compressed air is the *pneumatic network*. Increasing the accumulation capacities of its transport section provides temporary energy storage and availability to cover consumption peaks. In order to calculate the accumulation capacity of a transport section, the pressure and mass-flow variation laws for it have to be known. For this purpose the balance equation for nonstationary fluid flow through long ducts will be applied, and will be solved with the help of an approximate method suggested by I. A. Ciarnîi (6).

The momentum conservation equation is written in the form

$$\frac{\partial w}{\partial \tau} + w\frac{\partial w}{\partial x} = g_x - \frac{1}{\rho}\frac{\partial p}{\partial x} - \frac{\lambda}{d}\frac{w^2}{2} \tag{32}$$

where

w = flow velocity
τ = time
x = spatial coordinate
g = gravitational acceleration
ρ = density
p = pressure
λ = fluid-dynamic resistance ratio
D = duct internal diameter;
δ = boundary-layer thickness of flow section
$d = D{-}2\delta$. The fluid load loss was approximated with the loss corresponding to the steady flow expressed by the Weissbach–Darcy relation.

For the continuity equation we use

$$\frac{\partial \rho}{\partial \tau} + \frac{\partial(\rho w)}{\partial x} = 0 \tag{33}$$

From Eqs. (32) and (33) we have

$$\frac{\partial(\rho w)}{\partial \tau} + \frac{\partial(\rho w^2)}{\partial x} = \rho g_x - \frac{\partial p}{\partial x} - \rho\frac{\lambda}{d}\frac{w^2}{2} \tag{34}$$

On neglecting the terms due to kinetic energy variation [$\partial(\rho w^2)/\partial x$] and gravitational forces (ρg_x) in comparison with the term due to friction [$\rho(\gamma/d)\, w^2/2$], Eq. (34) becomes

$$\frac{\partial(\rho w)}{\partial \tau} = -\frac{\partial p}{\partial x} - \rho\,\frac{\lambda}{d}\,\frac{w^2}{2} \tag{35}$$

Equation (35), together with the continuity equation in Eq. (33) and the state equation $\rho{=}\rho\,(p)$, forms a determinate system of equations.

Considering the definition of the isothermal compressibility ratio and Newton's relationship for the sound propagation velocity in fluids, we obtain for barotropic fluids

$$\frac{\partial \rho}{\partial \tau} = \frac{d\rho}{dp}\frac{\partial p}{\partial \tau} = \frac{1}{a^2}\frac{\partial p}{\partial \tau}$$

The equation of continuity will be written as

$$\frac{\partial(\rho w)}{\partial x} = -\frac{1}{a^2}\frac{\partial p}{\partial \tau} \tag{36}$$

We linearize the nonlinear equation in Eq. (35) by approximating the expression $\lambda w/2d$ with its average value $2c=(\lambda w/2d)_m$, and we use the relationship suggested by I. A. Ciarnîi (6), assuming a constant fluid-dynamic resistance coefficient λ in the expression for c, so that $2c=(\lambda/3d)\ (w_{initial}+w_{end})$. Then the equation becomes

$$\frac{\partial(\rho w)}{\partial \tau} = -\frac{\partial p}{\partial x} - 2c\rho w \tag{37}$$

For long ducts at times $\tau > L/a$ the term $\partial(\rho w)/\partial\tau$ may be neglected, and from Eqs. (36) and (37) the following system is obtained:

$$-\frac{\partial p}{\partial x} = 2c\rho w \tag{38}$$

$$\frac{\partial p}{\partial \tau} = a^2\frac{\partial(\rho w)}{\partial x} \tag{39}$$

Eliminating the product ρw from the obtained equation, differentiating Eq. (38) with respect to x, substituting the result in Eq. (39), and introducing the notation $b=a^2/2c$, we obtain

$$\frac{\partial p}{\partial \tau} = b\frac{\partial^2 p}{\partial x^2} \tag{40}$$

The same technique is used to remove the pressure from Eqs. (38) and (39). Differentiating Eq. (38) with respect to time and Eq. (39) with respect to x, and equating them, we obtain

$$\frac{\partial(\rho w)}{\partial \tau} = b\frac{\partial^2(\rho w)}{\partial x^2} \tag{41}$$

Solving Eqs. (40) and (41) allows the pressure and mass-flow variation in nonstationary fluid flow through a long duct to be determined.

The pressure variation law can be obtained by applying the Fourier transformation (7):

$$p = \frac{2}{L}\sum_{n=1}^{\infty} e^{-(bn^2\pi^2/L^2)\tau}\left(\int_0^L f(x)\,\sin\frac{n\pi x}{L}\,dx + \frac{bn\pi}{L}\int_0^\tau \left[F_1(\tau) - (-1)^n F_2(\tau)\right] e^{(bn^2\pi^2/L^2)\tau}\,d\tau\right)\sin\frac{n\pi x}{L} \tag{42}$$

For the mass-flow variation law a relationship having the same form is found:

$$M = \frac{\pi d^2}{4}\rho w = \frac{\pi d^2}{2L}\sum_{n=1}^{\infty} e^{-(bn^2\pi^2/L^2)\tau}\left(\int_0^L g(x)\,\sin\frac{n\pi x}{L}\,dx + \frac{bn\pi}{L}\int_0^\tau \left[G_1(\tau) - (-1)^n G_2(\tau)\right] e^{(bn^2\pi^2/L^2)\tau}\,d\tau\right)\sin\frac{n\pi x}{L} \tag{43}$$

with the conditions

$$\tau = 0, \qquad \rho w = g(x) \tag{44}$$

$$x = 0, \qquad \rho w = G_1(\tau) \tag{45}$$

$$x = L, \qquad \rho w = G_2(\tau) \tag{46}$$

The functions $f(x)$, $g(x)$, $F_1(\tau)$, $F_2(\tau)$, $G_1(\tau)$, and $G_2(\tau)$ are calculated using regressions based on sets of experimental data.

The pressure and mass-flow values at different points of a pneumatic network are calculated with Eqs. (42) and (43) using numerical methods of integration. We use the following notation for experimentally determined pressures: p_s=adjusted pressure at buffer reservoir discharge valve p_n=compressed air pressure behind buffer reservoir in normal functioning regime (the flow supplied by the compressor meets the requirements of the consumers) p_d= compressed air pressure behind buffer reservoir when the flow supplied by the compressor is less than the flow required by consumers at the downstream end of the transport sections p_{cs}=compressed air pressure corresponding to pressure p_s p_{cn}=compressed air pressure corresponding to pressure p_n p_{cd}=compressed air pressure corresponding to pressure p_d

When the conditions

$$p_s \geq p(0,\tau) > p_n \quad \text{and} \quad p_{cs} \geq p(L,\tau) > p_{cn}$$

are met, the trunk line operates in accumulation regime. When

$$p(0,\tau) \geq p_n \quad \text{and} \quad p(L,\tau) \geq p_{cn}$$

the trunk line compensates for the extra flow rate required by the consumers. When

$$p(0,\tau) < p_{\mathrm{n}} \quad \text{and} \quad p(L,\tau) < p_{\mathrm{cn}}$$

the flow required by the consumers exceeds the flow supplied by the compressor and a second compressor must be coupled to the network to provide the required supplementary flow. The accumulated flow, compensated flow, and additional flow required are determined by Eq. (43) from the differences $\Delta M = M(0,\tau) - M(L,\tau)$ corresponding to each operating regime of the trunk line.

Possibilities for Compressed Air Storage with the Help of a Collector–Distributor Section. Pneumatic equipment is supplied with compressed air from a pneumatic distribution network at points relatively close to one another. Nevertheless, the distribution network is connected to two trunk lines coming from different compressor stations, to provide better service to pneumatic consumers. For large systems it is advisable to transform the pneumatic network from a branched network into an annular one, by developing an annular collector–distributor made up of one or several loops resulting from parallel connection of distribution ducts.

The collector–distributor will be supplied at the two ends of the arragement. Flexible supply is thus assured, leveling the load curve, consumption peaks being taken over by the collector–distributor using compressed air accumulated during low consumption periods. Supplying consumers with required flows at necessary pressures by this means involves certain restrictions on the pressure variations in the collector–distributor due to uneven consumption.

At the branching points of the consumers' supply ducts to the collector–distributor, the pressure must be higher than a value $p_{2\min}$ determined by the pressure required for pneumatic equipment and the losses accompanying compressed air flow along the distribution ducts. Collector–distributor supply points will be at a lower pressure than the value $p_{1\max}$ set at the buffer reservoir discharge valve in the ecompressor station.

The compressed air required for pneumatic equipment loads must be supplied over a 24 h cycle by meeting the following conditions:

$$p_1(L,\tau) < p_{1\,\max} \quad \text{and} \quad p_2(L,\tau) < p_{2\,\min} \qquad (47)$$

where

$p_1(L,\tau)$ = compressed air pressure variation at the branching points of the collector–distributor from trunk lines

p_2 = compressed air pressure variation at consumer–distributor consumption points

Two types of issues are raised in practice:

- Calculation of the needed geometrical size of the collector–distributor to cover variations in consumption
- Determination of time variations of flow and pressure at the supply points of the collector–distributor as functions of their variations at the consumption points, for certain geometrical sizes of the collector–distributor.

The required volume of the collector–distributor results from the condition of compensating momentary flow deficiencies during a 24 h period:

$$V = \frac{\sum Q_{\mathrm{di}}\tau_{\mathrm{dt}}}{p_2 T_{\mathrm{N}}/p_{\mathrm{N}} T_2 - p_1 T_{\mathrm{N}}/p_{\mathrm{N}} T_1} \qquad (m^3) \qquad (48)$$

where

$Q_{di}=Q_{ci}+\Delta Q_i-Q_{sci}$ = flow deficit during the time τ_{di} ($N\cdot m^3$/min)
Q_{ci} = flow required by consumers during the time τ_{di} ($N\cdot m^3$/min)
Q_{sci} = flow conveyed to compressor station during the time τ_{di} ($N\cdot m^3$/min)
τ_{di} = life span of flow deficit Q_{di} (min)
p_1, p_2, p_N = absolute pressures of compressed air before and after filling up the collector–distributor, and pressure corresponding to the normal physical state (bars)
T_1, T_2, T_N = absolute temperatures of compressed air before and after filling up the collector–distributor, and absolute temperature corresponding to the normal physical state (K)

The time taken to fill up the collector–distributor is calculated with the formula:

$$\tau_u = \frac{V}{Q_u}\left(\frac{p_2 T_N}{p_N T_2} - \frac{p_1 T_N}{p_N T_1}\right) \qquad \text{(min)} \tag{49}$$

where $Q_u=Q_{sc}-(\Delta Q+Q_c)$ is the available flow for filling ($N\cdot m^3$/min).

The flow and pressure variations in time in the collector–distributor supply are determined by Eqs. (42) and (43) when their variations at the consumption points are known.

Turning the collector–distributor into a pneumatic-energy “flywheel” provides the following benefits in the operation of pneumatic installations:

- It facilitates adjusting the pressure and flow of compressed air conveyed to consumers.
- It allows pneumatic energy storage so that energy can be supplied at the right moment;
- It levels the pneumatic installation load curve, providing for consumption peaks, so that no additional compressor has to be coupled to the network.
- Compressor stations may be sized to required average loads.
- It provides a stationary working regime for compressor stations.
- It increases the fluid stability of the pneumatic network.
- It represents a pneumatic energy supply in case of emergency at the compressor stations.
- It removes humidity from the compressed air.

Conclusion

The evolution of mankind is currently focused on harmonization with nature. Sustainable development will thus be a goal, especially in the energy sector, and energy costs will include externalities as well. In view of ecological restrictions and externality costs, some primary energy sources will become very expansive, and wind and solar energy will gain ground. Except for local utilization of wind or solar energy, considering the abundance, degree of dispersion, and intermittence of these sources, energy storage solutions must be found.

The favorable characteristics of storage systems based on compressed air and the necessity of providing autonomy in time for wind and solar energy conversion justify efforts at development and proliferation of energy storage by compressed air.

Because compressed air is an medium of energy transmission that is capable of simultaneously accumulating thermal and mechanical energy, a possible combination of wind and solar systems might be developed, using compressed air both as an energy transfer medium and as an energy storage medium.

BIBLIOGRAPHY

1. A. Bacu A. A. Bacu *Indrumator pentru Utilizarea Aerului Comprimat in Minerit (Guide to Using Compressed Air in Mining Engineering)*, Bucharest: Technica, 1972.
2. P. Zaugg Z. S. Stys H. Hoffeins Air storage power plants with special consideration of USA–conditions, presented at *11th World Conf. of Energy*, München, 1980.
3. A. Bejan *Advanced Engineering Thermodynamics*, New York: Wiley, 1988.
4. V. Radcenco *Criterii de Optimizare a Proceselor Termice (Optimization Criteria of Thermal Processes)*, Bucharest: Tehnica, 1977.
5. A. Bejan Method of entropy generation minimization, or modeling and optimization based on combined heat transfer and thermodynamics, *Rev. Gén. Therm.*, **35**: 637–646, 1996.
6. H. Pascal *Curgerea Nestaţionară a Gazelor prin Conducte Magistrale (Unsteady Gas Flow through Central Ducts)*, Bucharest: Academy Publishing House, 1967.
7. I. I. Irimie I. Matei *Gazodinamica Reţelelor Pneumatice (The Gasdynamics of Pneumatic Networks)*, Bucharest: Tehnica Publishing House, 1994.

READING LIST

B. Andersen Finite-time termodynamics and thermo-dynamic length, *Rev. Gen. Therm.*, **35**: 647–650, 1996.

S. Balkan Wärmespeicherantriebe für einen Schadstoffreien Fahrzeugbetrieb, Dissertation an der TU Berlin, 1974.

N. Bradeanu Instalatii Pneumatice Miniere (Mining Pneumatic Installations), Bucharest: Tehnica, 1976.

H. Bruntland *Our Common Future*, New York: Oxford University Press, 1987.

P. Chesse J. F. Hetet J. P. Frayret Determination par simulation d'une zone critique des champs compresseur située entre le pompage et la stabilité, *Entropie*, **201**: 9–11, 1996.

D. Coiante *Uno Sguardo al futuro dell'Energia, Studi e Ricerche*, ENEA, Italy: 1996.

Al. Danescu (ed.), *Termotehnica si Masini Termice (Thermotechnics and Heat Engines)*, Bucharest: Didactic and Pedagogic Publishing House, 1985.

K. Euler *Methods d'Accumulation de l'Energie Electrique*, Francfort sur le Main: Revue VARTA AG, 1984.

V. I. Evenko An exergetic appraisal of the thermodynamic perfection of compressors, *Thermal Eng.*, **44 (3)**: 233–238, 1997.

C. Flavin N. Lensen *Power Surge*, London: Norton, 1994.

S. Frandsen On the wind speed reduction in the center of large clusters of wind turbines, presented at EWEC' 91, Amsterdam.

P. Glansdorff I. Prigogine *Structure, Stabilité et Fluctuation*, Paris: Masson, 1971.

A. Houberechts *Thermodynamique Technique*, Paris: Dunod, 1962.

V. Ilie *Utilizarea Energiei Vântului (Utilization of Wind Energy)*, Bucharest: Tehnica, 1984.

I. I. Irimie S. Irimie I. C. Tulbure Solutions for turning to account the thermal energy of compressed air, Study, University of Petroşani, 1994.

M. F. Jischa *Herausforderung Zukunft*, Heildelberg: Spektrum Akademischer Verlag, 1993.

A. R. I. Kanal F. B. L. Janaina M. A. Zanardi Models of liquid storage tanks, *Energy*, **22 (8)**: 805–815, 1997.

D. Kanellopoulos Cluster efficiency model. An application to the seven wind farms of the HORS–QUOTA programme, *Proc. 1991 Thirteenth BWEA Wind Energy Conference*, 1991.

A. Kodra B. Fortunato A. Dadone Prestazioni e ottimizzazione di centrali eoliche, *Energia Mater. Prime*, **121**: 10–15, 1996.

A. Leca *Principles of Energy Management*, Bucharest: Tehnica, 1997.

T. Oroveanu (ed.) *Colectarea, Transportul, Depozitarea şi Distribuţia Gazelor (Collection, Transport, Storage, and Distribution of Gases)*, Bucharest: Didactic and Pedagogic Publishing House, 1985.

J. Schröder K. Gawron Latent heat storage *Energy Res.*, **5**: 103–109, 1981.

T. Svenson et al. *Environmental Consequences of New Energy Technology*, Swedish Council for Building Research, 1991.

I. C. Tulbure *Zustandsbeschreibung und Dynamik Umweltrelevanter Systeme*, Clausthal–Zellerfeld: CUTEC-Schriftenreihe, No. 25, 1997.

P. Zaugg Z. S. Stys Centrales avec réservoir d'air, specialement pour les conditions US, *Rev. Brown Boveri 12*, **67**: 723–733, 1980.

IOAN-IULIAN IRIMIE
SABINA IRIMIE
ILDIKO-CAMELIA TULBURE
University of Petrosani