UDC 536.7:621.311 DOI:10.30724/1998-9903-2019-21-3-4-27-37

# ASSESMENT OF COMPRESSED AIR ENERGY POTENTIAL FOR DETERMING EFFICIENCY OF AN AIR STORAGE POWER PLANT

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***Abstract****: The article is devoted to thermodynamic analysis of the processes occurring in energy storages of compressed atmospheric air. As a tool for study of compression-expansion processes, we propose a method of compiling and analyzing of exergy balance. As a result, the main characteristics can be assessed, namely, potential of compressed air and efficiency of all elements of the circuit and air storage power plant (ASPP) as a whole. The study procedure is shown on the example of an adiabatic ASPP. The dependences of the final temperature, specific exergy and compression work per 1 ths. $m^3$ of atmospheric air on the degree of its compression were calculated. Due to external factors ASPP of this type cannot have high thermodynamic efficiency. However, it has a number of economic advantages, and is well compatible with renewable energy sources, for example, with wind power plants. To increase the efficiency of the process, it is necessary to convert the ASPP type to non-adiabatic one, which will require changes in circuit and operation modes of its elements. At the same time, compression in the compressor should be carried out with the highest possible heat removal, and the expansion process in open-type gas turbine units should be carried out with heat supply.*

***Keywords****: energy storages, air storage power plant, thermodynamics analysis, exergy method.*

**Acknowledgments:** *The work was done as a personal initiative.*

**For citation:** Stepanov V.S., Stepanova T.B., Starikova N.V. Evaluaation the energy potential of compressed air for determining efficiencty of air-accumulating electric. *Proceedings of the higher educational institutions. ENERGY SECTOR PROBLEMS*. 2019; 21 (3-4):27-37. (In Russ). doi:10.30724/1998-9903-2019-21-3-4-27-37.

## Introduction

In recent years, many countries have significantly increased their interest in renewable energy sources (RES), the use of which does not violate the natural energy balance of our planet. Renewable energy sources include reserves that are replenished naturally and are practically inexhaustible in the foreseeable future. First of all, these include solar radiation, wind energy, river energy, etc. [1–3].

However, almost all sources of renewable energy have a probabilistic mode of energy output during a day, season, time period, which is a distinctive feature of all natural processes. Moreover, schedule of such power generation does not match with requirements of power system in terms of covering the load to consumers, and even is often opposed to it. In other words, these sources do not have the constancy of the output power, are not subject to regulation, and cannot adjust to the power system's power consumption mode. In other words, the use of RES is associated with a number of difficulties, due to the probabilistic nature of generation and, therefore, the complexity of integrating them into the operating mode of traditional power systems.

Currently, a new concept is actively being promoted - the construction of future smart energy systems, called the Smart Grid. This concept is aimed at achievement the following important goals: widespread use of modern means of control and management, including Internet technologies, integration of renewable energy sources into the power supply system, as well as various kinds of energy storage devices in order to increase the reliability and efficiency of power generation and its quality.

It is known that energy storage can be carried out in the form of potential energy of an agent (energy carrier), for example: water, by using it at water storage power plant (WSPP), or compressed gas (air), feeding it into compressionless gas turbines installed at the air storage power plants (ASPP) of various types.

Energy of compressed air can be accumulated by pumping it with a high-pressure compressor into specially designed tanks or underground natural and artificial containers. When required, compressed air from the storage is fed to the corresponding element of the ASPP.

There are two types of compressed air storages which differ by location of the storage. The first one is called the *ground storage system*, and the other is *underground storage system*. Each of them has its own specific requirements and scope. Currently, it is economically feasible to use underground tanks of compressed air as part of air-storage power plants with a capacity of 100 kW and more, and ground storage facilities can operate as part of small power plants with a capacity of 10 to 100 kW.

Natural gas pipelines, for whatever reason turned out to be idle can be used as ground storages of compressed air. Specially constructed tanks for storing compressed air can be used as part of a small capacity ASPP.

Underground compressed air storages are more economical, for which it is advisable to use natural voids in the earth's crust, salt caves, spent natural gas deposits, etc.

Energy storages of compressed air (ESCA) allow one to save large quantities (volumes) of energy for a long time at a relatively low cost. Therefore, their widespread use in the smart grid of the future (Smart Grid) has very good prospects in view of further improving the energy efficiency of the storages themselves.

Improving the processes of compression and expansion of air to increase the efficiency of ESCA is possible only when one understands the complex processes that occur in compressors, and the nature of changes in air parameters and energy transformations in other ASPP elements [1].

**Thermodynamic principles of air compression and expansion [4, 5]**

To describe the nature of change in air parameters in this process, the thermodynamic function of polytropes $pV^n = \text{idem}$ with different values of its exponent *n* is used.

All the variety of processes of compression and expansion of air can be divided into two types. The processes carried out without exchange of heat of a compressible/expanding agent with environment, that is for $dq=0$ and $dS=0$, are called *adiabatic* or *isentropic*. The polytropic index of such processes is $n=k$, where $k$ is adiabatic index equal to the ratio between the isobaric and isochoric heat capacities of air, $k = \dfrac{C_p}{C_V}$. All other processes occuring with heat exchange are called non-adiabatic or diabatic.

The processes of air compression can be carried out with heat removal from the compressible agent, the polytropic index for them is $n<k$. Moreover, if the process of gas compression takes place with complete heat removal, i.e. the process proceeds at a constant initial temperature $T_1 = \text{idem}$, then it is called *isothermal*. The polytropic index of such processes is $n$=1.

In general case, for non-adiabatic air compression processes carried out with heat removal, the polytropic index is $n<k$, and it is $n>k$ when heat is supplied. The polytropic indices for non-adiabatic expansion processes when heat is applied are $n<k$, and $n>k$ when heat is removed.

Fig. 1 shows the relationship between the agent state parameters in an ideal compressor and the polytropic index, describing various processes of air compression. The diagram shows the process of air compression in a reciprocating compressor with perfect valves, without dead space. The work consumed by compressor in one cycle is equal to the area under the curve described by the polytrope with the corresponding index *n*.

As it is seen from Fig. 1, this work has a minimum value during air compression process with complete removal of heat from it, which is described by the isotherm with polytropic index $n = 1$ (area *1-2-3-4-1*). Such a process is almost impossible to carry out.

Also it is impossible to carry out the adiabatic compression process with the polytropic index $n=k$, since it is not possible to create a compressor in which there would be no heat exchange between the compressible agent and the environment. The work of adiabatic air compression (area *1-2′-3-4-1*) is more than the work of isothermal compression.

Air compression processes which can be really carried out, are non-adiabatic (diabatic) with polytropic indices in the regions $1<n<k$ and $n>k$. Therefore, it is not surprising that adiabatic gas compression/expansion process is taken as the reference, and the characteristics of the compressible agent are used as the limiting ones.

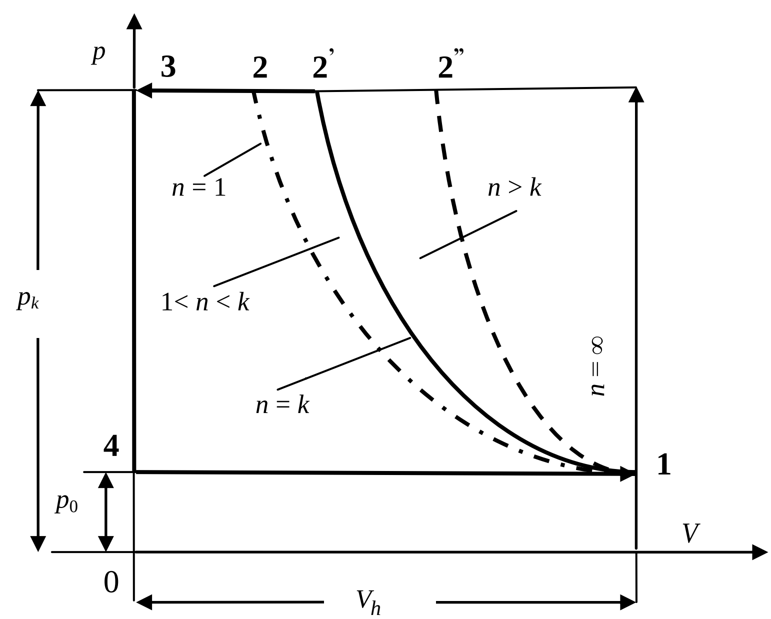


Fig. 1. Diagram of air compression processes in *p-V* coordinates

We consider the process diagram of an ideal compressor, shown in fig. 1. The process of air intake *1-4* occurs at constant pressure $p_0 = \mathrm{idem}$. The compression process is carried out according to the reversible adiabat *1-2*, described by the equation $pV^k = \mathrm{idem}$. The process of air ejection *2-3* occurs at constant pressure $p_k = \mathrm{idem}$. The work spent by the compressor for the cycle is equal to the area *1-2-3-4-1*. It characterizes the thermomechanical exergy of compressed air and can be determined using the formula

$$l = e_{\mathrm{TM}} = \int_1^2 V\,dp\,, \qquad (1)$$

where $V$ is cylinder volume; $dp$ is infinitely small change in pressure.

From the adiabatic equation $p_1 V_1^k = p\,V^k$ it follows:

$$V^k = V_1 \frac{p_1^{1/k}}{p^{1/k}} . \quad (2)$$

As a result of substitution and integration, we obtain an expression for determining the work of adiabatic air compression:

$$l = e_{\text{TM}} = \frac{k}{k-1} \, p_1 V_1 \left[ \left( \frac{p_2}{p_1} \right)^{\frac{k-1}{k}} - 1 \right]. \quad (3)$$

Thus, in systems of storage and use, compressed air is constantly subjected to processes of compression and expansion. However, the specialists have some doubts in regard to use of the terms of energy or exergy while answering the following questions: What kind of energy transformations occur with air? How its energy potential changes? And how one can evaluate this potential? This article uses the concepts of exergy and exergy balance [6–9].

The equation of energy and exergy balances of reversible processes of compression and expansion for a unit mass of any gas is:

$$l = h_2 - h_1 + q \quad (4)$$

and

$$l = e_2 - e_1 + e_q = e_2 - e_1 + q \cdot \tau_e . \quad (5)$$

Here $\tau_e = \dfrac{T - T_0}{T}$ is heat efficiency coefficient, where $T$ is an arbitrary temperature; $T_0$ is ambient temperature.

Processes can be carried out at different ratios of temperatures $T$ and $T_0$. Of interest is the case of compression at $T = T_0$, when $e_q = 0$. In this case $l = e_2 - e_1$, that is, the work of isothermal compression (expansion) at ambient temperature is equal to the difference between gas exergy in the initial and final states.

From equation (4) it follows that the change in gas energy (in a continuous process, gas energy is measured by enthalpy) is equal to the difference between the spent work $l$ and the heat removed $q$:

$$h_2 - h_1 = l - q . \quad (6)$$

For an ideal gas, this value is zero, because its exergy does not depend on pressure, and the heat released during the compression process is exactly equal to the spent work.

In general case, for a real gas $q \neq l$, and the difference $h_2 - h_1$ is determined by the magnitude and sign of the isothermal Joule-Thomson effect of the gas under consideration $\Delta h_{\text{T}}$ at a given temperature, i.e.

$$\Delta h_{\text{T}} = h_1 - h_2 . \quad (7)$$

The magnitude and sign of the Joule – Thomson effect is determined by the ratio between the gas work and the work of external pressure forces, as well as the properties of the gas itself, in particular the size of its molecules and their interaction.

For gases such as air, oxygen, and nitrogen at $T_0$ and pressures up to 30 MPa, $\Delta h_{\text{T}}$ are greater than zero (positive Joule – Thomson effect) and, therefore, the energy of such compressed gases is less than for uncompressed ones ($h_2 < h_1$).

For other gases (helium, neon, hydrogen) at $T_0$ $\Delta h_{\text{T}} < 0$ (negative Joule – Thomson effect), and the energy of compressed gas is several percent more than that of the expanded one. However, this difference is very small compared with the amount of work spent for gas compression.

Thus, the energy of compressed gas is slightly different from the energy of uncompressed gas, and in most cases is less than it.

In cases when isothermal compression occurs at $T \neq T_0$ the value $e_q$ in equation (4) is not zero.

The energy balance of the compression process in general can be written as:

$$l = q - \Delta h_{\mathrm{T}} . \tag{8}$$

And the exergy balance has the following form

$$l = e_2 - e_1 + e_q = \Delta e + q \cdot \tau_e . \tag{9}$$

By substituting $q$ from (8) into (9) and using the expression for $\tau_e$, we obtain:

$$l = \Delta e \frac{T}{T_0} + \Delta h_{\mathrm{T}} \left( \frac{T}{T_0} - 1 \right) = ( \Delta e + \Delta h_{\mathrm{T}} ) \frac{T}{T_0} - \Delta h_{\mathrm{T}} \tag{10}$$

and

$$q = ( \Delta e + \Delta h_{\mathrm{T}} ) \frac{T}{T_0} . \tag{11}$$

For an ideal gas, the value of the isothermal Joule-Thomson effect $\Delta h_{\mathrm{T}} = 0$. In this case, expressions (10) and (11) are reduced to the following formula:

$$l = q = \Delta e \frac{T}{T_0} . \tag{12}$$

In order to estimate the energy potential of compressed air at its various pressures, we performed calculations of thermomechanical exergy of 1000 m$^3$ of atmospheric air for the corresponding pressures. In the calculations it was assumed that exergy of air is equal to the work of its adiabatic compression in an ideal compressor, which was calculated using expression (3).

The results of calculations for pressure ranges from 3 to 90 atm are given in Table. 1. The resulting dependence can be used to solve a number of specific tasks, for example: to optimize the composition of virtual power plants using the energy of compressed air storages or when choosing the composition of generating sources of autonomous power systems and optimizing their operating modes, taking into account the uneven load curve of the considered system.

In addition, the thermomechanical exergy of compressed air allow us to determine the energy potential of air entering the storage facility at known pressures $p_1$ and $p_2$. For a known volume $V$, it can be found as the difference in exergies at corresponding pressures, i.e.

$$\Pi_{\mathrm{f}} = ( e_2 - e_1 ) \cdot V . \tag{13}$$

Obviously, the potential of the same air volume coming from the storage to the gas turbine will be the same. For example, if the pressure in the storage has changed from 80 to 90 atm. while the compressor delivers 100 ths. m$^3$ of air, then the energy potential of this air volume will be

$\Pi_f$ =(257,80–245,98) 100=1182 kW·h.

Since the energy potential of compressed air is compression work, then, knowing this value, it is possible to determine the amount of electrical energy spent on the compressor drive when it compresses air of a given volume. This value can be found using the expression:

$$W_{el}^{c} = \frac{\Pi_f}{\eta_c \cdot \eta_{ed}} , \tag{14}$$

where $\eta_c$ is efficiency of compressor; $\eta_{ed}$ is efficiency of electric drive.

Similarly, one can determine the amount of electrical energy that can be obtained using the gas turbine unit:

$$W_{el}^{c} = \Pi_f \cdot \eta_{GT} \cdot \eta_{eg} ,$$

where $\eta_{GT}$ is efficiency of gas turbine unit; $\eta_{eg}$ is efficiency of electric generator.

The obtained values of the thermomechanical exergy of air can be used to estimate the energy potential of compressed air of storages used in various circuits and in different operating modes of ASPP.

**ASPP schematics and hardware**

As was noted above, heat is generated during the compression phase, i.e. the compressed air is heated. The higher is the degree of increase in air pressure $\beta$, the higher its final temperature $T_{fin}$ is:

$$T_{fin} = T_0 \left[ \beta^{\frac{n-1}{n}} \right]. \tag{15}$$

Two typical ASPP schemes are distinguished depending on the use of heat generated during air compression: adiabatic and non-adiabatic. Further we consider the features of building an adiabatic ASPP, the scheme of which is shown in Fig. 2

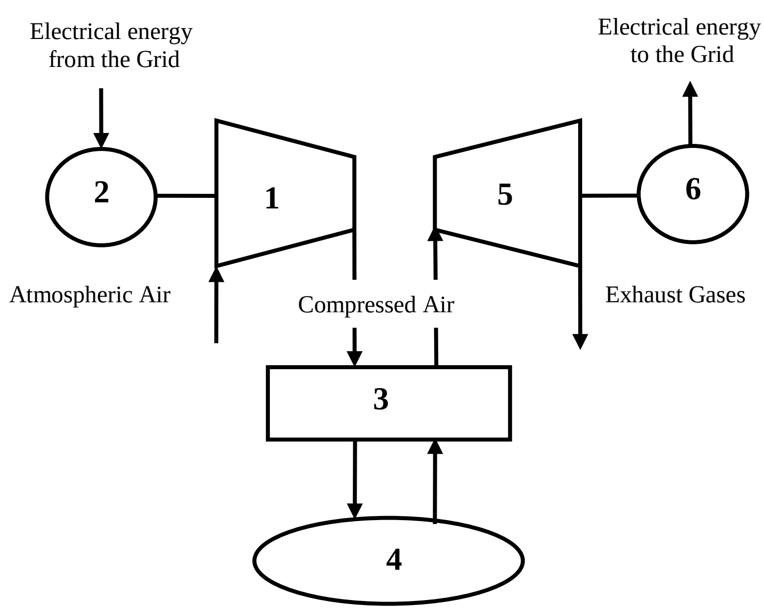


Fig. 2. Schematic diagram of the adiabatic ASPP: *1* – Compressor; *2* – Electric drive; *3* – Heat accumulator – heat exchanger; *4* – Compressed air storage; *5* – Uncompressed gas turbine (UGT); *6* – Electric generator

The ASPP includes: a compressed air storage unit, a compressor driven by an electric motor, an uncompressed gas turbine and an electric generator located on the same shaft as the turbine. The ASPP operation principle is as follows. The air compressed in the compressor, is adiabatically heated to the appropriate temperature, depending on the degree of increase in pressure $\beta$. As $\beta$ increases, the temperature of the compressed air leaving the compressor increases, which, in turn, leads to an increase in the work expended on the compressor drive. To obtain the dependence of the final temperature on the degree of air compression, calculations were made using formula (15).

Consequently, the increase in $T_{fin}$ leads to an increase in compression work, which was determined by formula (12). The results of these calculations are given in Table 1.

Table 1

Relationship between final temperature, specific exergy and compression work of 1 ths. $m^3$ of atmospheric air and the degree of its pressure increase

| $\beta = \frac{p_{fin}}{p_0}$ | Final temperature of the compressed air $T_{fin}$ | Thermomechanical exergy $e_{TM}$ | | Work for compression $l_{com}$ | |
|---|---|---|---|---|---|
| | | kJ | kW·h | kJ | kW·h |
| 3 | 401,24 | 130,44 | 36,21 | 178,50 | 49,58 |
| 5 | 464,23 | 206,44 | 57,22 | 327,00 | 90,83 |
| 10 | 565,97 | 330,88 | 91,90 | 639,25 | 177,58 |
| 20 | 689,43 | 478,29 | 132,86 | 1126,36 | 312,88 |
| 30 | 774,11 | 580,45 | 161,24 | 1533,54 | 425,92 |
| 40 | 840,62 | 660,69 | 183,52 | 1896,00 | 526,80 |
| 50 | 895,91 | 727,50 | 202,08 | 2224,70 | 617,97 |
| 60 | 943,81 | 785,12 | 218,09 | 2529,63 | 702,69 |
| 70 | 986,31 | 836,38 | 232,39 | 2814,42 | 781,70 |
| 80 | 1024,66 | 882,69 | 245,13 | 3085,0 | 856,94 |
| 90 | 1060,32 | 925,11 | 256,98 | 3349,82 | 930,30 |

The obtained results allow one to determine the energy potential of compressed air, and therefore make it possible to compose the exergy balance of the corresponding circuit elements and the ASPP as a whole and determine their efficiency.

In the considered scheme, in order to increase the efficiency of a compressed air energy storage device, it is usually envisaged to utilize its heat after compression using a heat exchanger-accumulator, through which compressed air passes before it enters the storage unit. Compressed air from the storage is supplied to the uncompressed turbine being previously pre-heated by the heat stored in heat exchanger. However, in this scheme, the heat of compressed air entering the turbine may have a relatively low temperature, much lower than $T_c$. Therefore, the efficiency of such ASPP will not be very high. This does not mean that the use of adiabatic ASPP never makes sense. As it can be seen from Fig. 2, such a scheme is simple in terms of a set of equipment (there is no compressor for a gas turbine), and therefore does not require large capital investments, i.e. it has economic advantages.

The experience of usage of gas utilization non-compressor turbines (GUNT) operating on blast-furnace gas overpressure at the metallurgical plants convincingly shows their technical and economic efficiency. Specific capital investments per 1 kW of installed capacity of a gas-expanding station (GES) with GUNT are 35 % less, and operating costs for generating 1 kW·h of electricity for it are 45 % less compared with that for CHP [9, 10].

So, according to the facts presented in the theoretical section of the paper, in order to increase the energy efficiency of adiabatic ASPP, it is necessary to fundamentally change its scheme. Compression in compressor should be carried out with the maximum possible removal of heat (bringing the process closer to isothermal), and the expansion process in turbine should be carried out with heat supply (bringing the process closer to the adiabatic one).

A schematic diagram of such a non-adiabatic ASPP is shown in Fig. 3.

The scheme shows that compressor is equipped with coolers to remove the heat released during air compression. To increase the efficiency of using compressed air energy, it is directed not into the uncompressed gas turbine, but into the scheme of a conventional open-type gas turbine unit (GTU). Such GTU consists of a fuel combustion chamber (gaseous or liquid), the combustion products of which are sent to a gas turbine (GT). GT shaft is connected to shaft of compressor and electric generator. For fuel combustion, compressed air is supplied to the combustion chamber from its own compressor. In general case, a gas turbine unit can be of any scheme and operate in autonomous mode, in particular, in cogeneration mode.

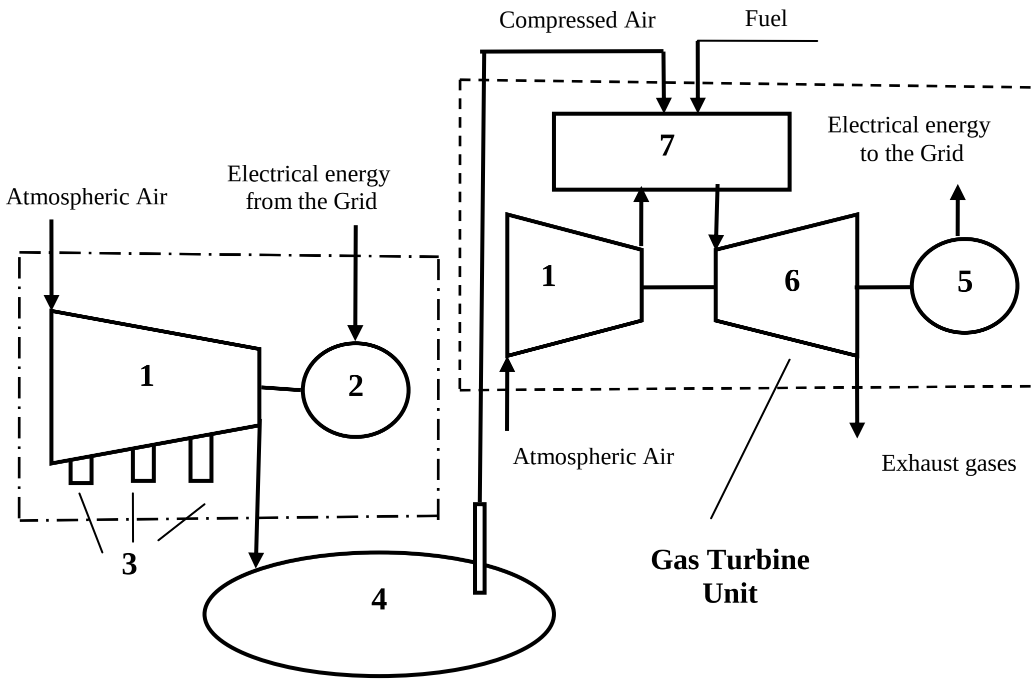


Fig. 3. Schematic diagram of a non-adiabatic ASPP in combination with GTU:
*1* – Compressor; *2* – Electric motor; *3* – Coolers; *4* – Compressed air storage;
*5* – Generator: *6* – Turbine: *7* – Combustion chamber

When considering GTU combination with a compressed air energy storage unit, compressed air from the storage is fed into the fuel combustion chamber, replacing the air and work on its compression in its own compressor of the gas turbine station. As a result, the output of GTU significantly increases. The efficiency of such a combination of ASPP + GTU can be increased by using the heat of gases coming out of the turbine (about 600–700 °C). This heat can be used to heat the compressed air before it is fed into the combustion chamber, which will reduce the amount of burned fuel and, consequently, reduce the negative impact of the object on the environment.

There is another way of using this heat by organizing production of by-products, i.e. hot water or steam at GTU (cogeneration mode). It also leads to an increase in energy efficiency of the complex and a reduction in harmful emissions to the environment.

The non-adiabatic (diabatic) ASPP with a capacity of 290 MW has been operating according to the scheme (shown in Fig. 3) in Germany since 1978. It has the following characteristics: the phase of air pumping into the storage facility lasts 8 hours; to compress 300 ths. $m^3$ of air from 46 to 72 atm, a compressor with a 60 MW electric motor is used. During the discharge phase, the unit provides the system with a capacity of 290 MW for 2 hours. At the same time, for each output of 1 kWh of electricity, it additionally consumes 0,8 kWh of electricity and 1,6 kWh of natural gas energy.

The authors tried to evaluate the complex efficiency using its known characteristics, which, however are insufficient to compile its full energy balance, developed using the 1st and 2nd principles of thermodynamics [6, 8]. The amount of accumulated exergy (potential) of compressed air was determined by the method presented above. As it can be seen from Table. 2, the achieved efficiency of the complex was $\eta_{compl} = 29{,}6\%$.

Table 2

The main energy characteristics of the complex combining non-adiabatic ASPP and gas turbine installation (real scheme)

| Consumes | Generates |
|---|---|
| 1. Compressor drive *(ED) $W_c = \frac{N_d}{\eta_d \eta_c} t_z = \frac{60}{0{,}93 \cdot 0{,}85} \cdot 8 = 607{,}6\ MW \cdot h$ | 1*. Electrical energy generation into the system $W_{gen} = P_m \cdot t_{dis} = 290 \cdot 2 = 580\ \text{MW} \cdot \text{h}$ |
| 2. Additional consumption of electrical energy $W_{add} = W_{gen} \cdot 0{,}8 = 580 \cdot 0{,}8 = 464\ \text{MW} \cdot \text{h}$ | 2*. The accumulated exergy (potential) of compressed air $\Pi_{comp} = (e_2 - e_1) \cdot V_a = (235 - 195) \cdot 300 = 12\ \text{MW} \cdot \text{h}$ |
| 3. Additional consumption of natural gas $W_{add\ gas з} = W_{gen} \cdot 1{,}6 = 928\ \text{MW} \cdot \text{h}$ | |
| **Total:** 1999,6 $\text{MW} \cdot \text{h}$ | $\eta_{compl} = \frac{592}{1999{,}6} = 0{,}296$ |

As noted above, the energy efficiency of the complex can be significantly increased if its structure is changed by replacing the compressor drive from electric motor with the drive from wind power plant (WPP), which will eliminate this item of electricity consumption from the network. In this case, the complex efficiency can achieve $\eta_{compl} = 42{,}5\%$ (Table 3).

As for the use of heat from GTU exhausted gases, there is no enough information to assess its impact on the complex efficiency. Using estimated calculation we determined the exergy of utilized gases heat, which amounted to $E_q = 23\ \text{MW} \cdot \text{h}$. This allows one to determine the complex efficiency, which can be achieved: $\eta_{compl} = \frac{592 + 23}{1392} = \frac{615}{1392} = 0{,}442.$

Table 3

The main energy characteristics of the complex combining non-adiabatic ASPP, GTU and wind power plant to increase its efficiency

| Consumes | Generates |
|---|---|
| 1. Compressor drive *(ED) | 1*. Electrical energy generation into the system $W_{gen} = P_m \cdot t_{dis} = 290 \cdot 2 = 580\ \text{MW} \cdot \text{h}$ |
| 2. Additional consumption of electrical energy gas $W_{add} = W_{gen} \cdot 0{,}8 = 580 \cdot 0{,}8 = 464\ \text{MW} \cdot \text{h}$ | 2*. The accumulated exergy (potential) of compressed air $\Pi_{comp} = (e_2 - e_1) \cdot V_a = (235 - 195) \cdot 300 = 12\ \text{MW} \cdot \text{h}$ |
| 3. Additional consumption of natural gas $W_{add\ gas} = 928\ \text{MW} \cdot \text{h}$ | |
| **Total:** 1392 $\text{MW} \cdot \text{h}$ | $\eta_{compl} = \frac{580 + 12}{1392} = \frac{592}{1392} = 0{,}425$ |

**Conclusions**

Compressed air energy storage devices are capable of storing a significant amount of potential energy that can be used in air-storage power plants. Depending on whether the heat of the compressible agent is discharged or not, ASPP is divided into two types - adiabatic and nonadiabatic. This article considers both types of ASPPs. The air in the compressor, storage and other elements of the ASPP is in compression-expansion modes. This means that it is necessary to be able to determine the potential of compressed/expanded air required to compile the exergy balance of the corresponding elements of the scheme and the ASPP as a whole, without which it is impossible to evaluate their efficiencies.

The article proposes the method for determining the energy potential of compressed air, the procedure for drawing up the exergy balance and determining the efficiency of the corresponding circuit elements. Based on the analysis of the processes of air compression (expansion) in the adiabatic ASPP, it was concluded that such a scheme cannot have high efficiency from the standpoint of thermodynamics. However, this type of ASPP has a number of economic advantages: simplicity, low investment, due to the absence of a compressor, and a combustion chamber for a gas turbine. To increase the process efficiency, it is necessary to convert the ASPP type to non-adiabatic, which will require changing the scheme and operation mode of its elements. At the same time, compression in the compressor should be carried out with the highest possible heat removal, and the expansion process in open-type gas turbines should be carried out with heat supply.

Both types can successfully work in conjunction with a wind power plant, which is able to replace the compressor motor, thereby reducing power consumption from the grid.

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***Received*** ***October ,16 2018***