Gas accumulator emptying process

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Abstract. Gas-charged accumulators serve as fundamental components in various technical systems, functioning both as primary units and supplementary safety devices. These systems may incorporate conventional pneumatic actuators or specialized actuator designs that form integrated mobile assemblies with the accumulator tank. This study develops and analyzes a thermodynamic model of gas behavior within accumulator tanks, with particular emphasis on heat exchange effects between the internal gas and external environment. A dimensionless criterion is derived from ideal gas thermodynamics to structurally and qualitatively evaluate these thermal interactions. The criterion enables rapid comparative analysis of system properties between real and ideal gas models. Certain applications may warrant development of analogous criteria for real gas models. The analysis employs technical measurement units to facilitate conversion to dimensionless parameters and process description. The proposed modeling approach provides a framework for evaluating thermal effects in gas-charged accumulator systems, with potential applications in design optimization and performance assessment

1 Introduction

Hermetically sealed containers filled with compressed gas (typically air) function as primary or auxiliary energy accumulators, providing motive power for system actuators, operational components, or emergency backup systems [1-3]. These compressed air reservoirs serve as autonomous energy sources capable of performing critical safety functions, particularly during power supply interruptions in industrial settings. The fire and explosion safety aspects of such pneumatic systems represent additional important considerations.

Gas-charged pneumatic systems find practical application through two fundamental operational principles. The first principle employs stationary actuators, exemplified by ground-based unmanned aerial vehicle launch systems utilizing pneumatic cylinder actuators [4], or spacecraft orientation systems incorporating rotary gas-dynamic actuators [5]. A specialized application involves emergency gas main shut-off valves activated during power failures, with particular relevance for remote pipeline locations where mainline gas may substitute tank-stored gas [6].

The second operational principle involves mobile tank systems where the pressurized vessel itself moves in conjunction with mounted payloads. In such configurations,

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environmental media (atmospheric or aquatic) may serve as reaction surfaces for thrust generation. While compressed air systems demonstrate lower specific impulse compared to rocket propulsion, their utility persists in niche applications requiring compact, self-contained actuation.

Effective implementation of gas-charged systems requires comprehensive characterization of tank thermodynamics, including parameter ranges during filling/emptying cycles and dimensional constraints. Temperature dynamics during decompression merit particular attention due to potential operational limitations, as evidenced by extensive research on hydrogen-based systems [7-8].

This study analyzes the decompression dynamics of high-pressure air tanks (modeled as ideal gas), incorporating tank geometry, initial pressure conditions, and wall heat transfer characteristics. The computational framework expresses heat flux and mass extraction rates through dimensionless ratios, enabling qualitative assessment of thermal effects on system performance. This approach facilitates comparative analysis between ideal and real gas behaviors while accommodating measurement uncertainties inherent in empirical parameter determination.

2 Energy of the tank emptying process

The equation for the energy balance in the emptying process of a constant-volume tank taking into account the effect of heat transfer is written as [9, 10]:

$$id\Theta + dQ = d(\Theta u), \tag{1}$$

where

i is the specific heat content of gas in the cavity, [kcal/kg], ($i = c_pT$);

c_p is the specific heat capacity at a constant pressure ($p = const$, [kcal/(kg · grad]);

Θ is the amount of gas in the cavity (mass) [kg];

u is the internal energy, [kcal/kg], ($u = c_vT$);

c_v is the specific heat capacity at a constant volume ($v = const$, [kcal/(kg · grad)];

dQ is the heat transfer between the gas (kcal) in the cavity and the environment, which can be represented by the relation $dQ = \alpha F^*(T - T_a)dt$;

$F^*$is the heat transfer surface between gas and accumulator walls, [m^2];

t is time, [sec];

T, T_a is the temperature of the air, respectively, in the cavity and the environment (absolute temperature), [K];

α is the heat transfer coefficient, [kcal/(m^2 · sec · grad)].

Substituting $i = c_pT$, $u = c_vT$, into equation (1), and excluding dT using the state equation, after simple transformations we obtain:

$$kRTd\Theta + \frac{R}{c_v}dQ = Vdp. \tag{2}$$

The change in the amount of gas in the tank is $d\Theta = -Gdt$, determined by the expression:

$$G = \frac{fBp\varphi(z)}{\sqrt{T}}, \quad \varphi(z) = \varphi\left(\frac{p_a}{p}\right) = \varphi^*, \quad B = \sqrt{\frac{2gk}{(k-1)R}},$$

where

G is the gas flow rate, [kg/sec];

$\varphi^* = 0.259$ is the supercritical flow function of the gas outflow;
p_a, p is the ambient pressure and the gas pressure in the tank, [kg$/m^2$];
f is the cross-sectional area of the outlet channel (m^2);
$k = 1.4$ is the adiabatic index;
$R = 29.3$ ([kg $\cdot$ m$/$(kcal $\cdot$ grad]) is the gas constant;
$V$is the gas volume $[m^3]$.

Assuming the heat transfer through the tank wall with the environment is convective and the heat transfer coefficient is constant, let us use the above expression of the heat transfer process, which together with dQ, $d\Theta$ and G we substitute into equation (2). As a result, we obtain:

$$\left[-kRTfB\frac{p}{\sqrt{T}}\varphi^* + \frac{(k-1)}{A}\alpha F(T_a - T)\right]dt = Vdp, \tag{3}$$

where $A = 0.002345$ is the thermal equivalent of work, [kcal/kg $\cdot$ m].

In order to simplify the model and analyze it further, we proceed to dimensionless parameters using the following relations between real and dimensionless variables:

$$t = q_1\tau, \quad p = q_2\sigma, \quad T = q_3\vartheta. \tag{4}$$

By substituting relations (4) into equation (3), we obtain:

$$\left[-kRBf\varphi^* q_2\sigma\sqrt{q_3\vartheta} + \frac{(k-1)}{A}\alpha FT_a(1-\vartheta)\right]q_1 d\tau = Vq_2 d\sigma,$$

or, choosing $q_2 = p_a$, $q_3 = T_a$, we have:

$$\left[-kRBf\varphi^* q_2\sigma\sqrt{T_a\vartheta} + \frac{(k-1)}{A}\alpha FT_a(1-\vartheta)\right]q_1 d\tau = Vq_2 d\sigma.$$

After dividing all terms of the equation by Vp_a and reducing them to dimensionless form, we have:

$$\left[\frac{-kRBf\varphi^*\sigma\sqrt{T_a\vartheta}}{V} + \frac{(k-1)\alpha FT_a(1-\vartheta)}{AVp_a}\right]q_1 = \frac{d\sigma}{d\tau},$$

or

$$-C_1\sigma\sqrt{\vartheta} + C_2(1-\vartheta) = \frac{d\sigma}{d\tau},$$

where

$$\frac{kRBf\varphi^*\sqrt{T_a}q_1}{V} = C_1, \qquad \frac{(k-1)\alpha FT_a(1-\vartheta)q_1}{AVp_a} = C_2.$$

To determine q_1, the value of the parameter C_1 should be set, for example, $C_1 = 1$. From here we get:

$$q_1 = \frac{V}{kRBf\varphi^*\sqrt{T_a}}; \quad C_2 = \frac{(k-1)\alpha F\sqrt{T_a}}{kARBf\varphi^* p_a}, \tag{5}$$

and the first basic equation:

$$-\sigma\sqrt{\vartheta}+C_2(1-\vartheta)=\dot{\sigma}, \tag{6}$$

If we change C_1 by some other constant value instead of one, then additional numerical factors will appear in the expressions for $C_1$and C_2. The second basic equation is obtained directly from the equation of state of the gas $pV=\Theta RT$.By transformation and differentiation; it is reduced to the form:

$$d\Theta=\frac{(dpT-pdT)V}{RT^2}, \tag{7}$$

where $d\Theta=-Gdt$.

After simple transformations, the equation (7) takes the form:

$$dT=\frac{Tdp}{p}+\frac{fB\sqrt{T_a}^3\varphi^*Rdt}{V},$$

and after the transition to dimensionless parameters taking into account the already accepted values q_1, q_2, q_3 we obtain:

$$T_a d\vartheta=\frac{T_a\vartheta d\sigma}{\sigma}+\frac{fB\sqrt{T_a}^3\varphi^*R\sqrt{\vartheta}^3q_1d\tau}{V}.$$

After dividing by T_a and rebuilding, we have:

$$d\vartheta=\frac{\vartheta d\sigma}{\sigma}+\frac{fB\sqrt{T_a}\varphi^*R\sqrt{\vartheta}^3q_1d\tau}{V}$$

or

$$C_3\sqrt{\vartheta}^3+\frac{d\sigma}{\sigma d\tau}=\frac{d\vartheta}{d\tau},$$

where $C_3=\frac{1}{k}$ is the result of substituting the previously found q_1 value.

$$\frac{fB\sqrt{T_a}\varphi^*R}{V}\cdot q_1=\frac{fB\sqrt{T_a}\varphi^*R}{V}\cdot\frac{V}{kRBf\varphi^*\sqrt{T_a}}=\frac{1}{k}.$$

As a result, an initial system of basic equations was obtained to study the process of changing the pressure and temperature of the gas in the tank over time:

$$\begin{gathered}-\sigma\sqrt{\vartheta}+C_2(1-\vartheta)=\dot{\sigma},\\ \frac{1}{k}\sqrt{\vartheta}^3+\frac{\vartheta\dot{\sigma}}{\sigma}=\dot{\vartheta}.\end{gathered} \tag{8}$$

Its solution is performed in the usual way: the values of the generalized dimensionless parameter C_2 and the initial values of dimensionless variables $\tau_0=0$, $\sigma_0=p_a/p_0$; $\vartheta_0=$

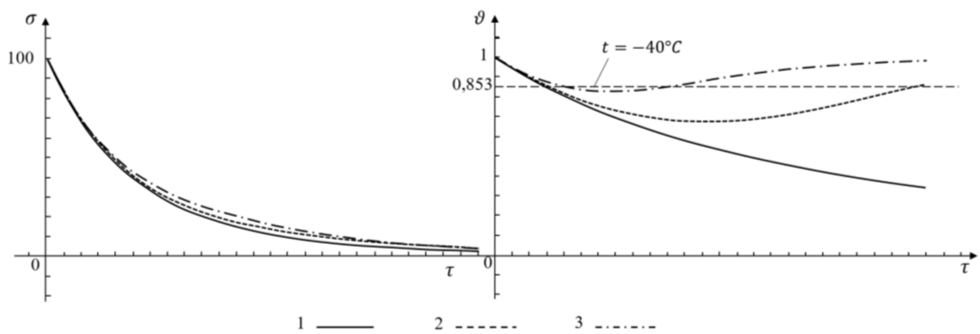
T/T_a are set. These data are sufficient to calculate the initial values of the rates of change of the variables $\dot{\sigma}_0$, $\dot{\vartheta}_0$ and numerically solve the system.

The single process criterion C_2 expresses the ratio of heat transfer intensities and gas flow rates in an energy form, which is characterized by a combination of several parameters.

An important factor is that the process characteristic does not depend on the tank volume, which determines the duration of the process. The criterion indirectly takes into account the influence of ambient temperature on the process.

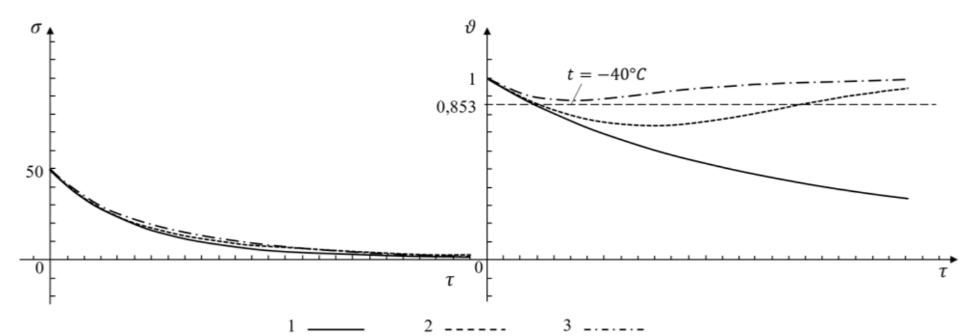
3 Modeling the gas outflow process from the tank

Figures 1 and 2 show families of curves characterizing in dimensionless form the processes of pressure and temperature changes of gas in the tank at $C_2 = 0.1$ and 5 ($C_2 = 0$ corresponds to complete absence of heat transfer) and at two initial conditions $-\sigma_0 = 100$, $\vartheta_0 = 1$ (which corresponds to $p_0 = 50$ MPa, $T_0 = 273K$) and $\sigma_0 = 50$, $\vartheta_0 = 1$ (i.e., $p_0 = 50$ MPa, $T_0 = 273K$).



(a) (b)

Fig. 1. The processes of changing the pressure σ and temperature ϑ of the gas in the tank under initial conditions: $\sigma_0 = 100, \vartheta_0 = 1(1 - C_2 = 0; 2 - C_2 = 10; 3 - C_2 = 50)$



(a) (b)

Fig. 2. The processes of changing the pressure σ and temperature ϑ of the gas in the tank under initial conditions: $\sigma_0 = 50, \vartheta_0 = 1(1 - C_2 = 0; 2 - C_2 = 10; 3 - C_2 = 50)$

The relationship between dimensional and dimensionless quantities is expressed by dependences (4). Since dimensionless values of pressure and temperature are expressed in scales $p_a = 1$ MPa and $T_a = 273$ K, the transition from dimensionless quantities of pressure and temperature to their real values is quite simple. As for time, $t = q_1 \tau$, where q_1 is represented by the complex expression (5), which includes a number of real system parameters. A similar remark can be made regarding the criterion C_2, which characterizes the relationship between the heat transfer and emptying processes.

According to the curves shown in Fig. 1, it is possible to estimate how low the air temperature may drop during the outflow process, if the pressure was high enough at the beginning of the process and the heat transfer was weak. So even at $C_2 = 100$ (this is relatively active heat transfer) and $p_0 = 100$ MPa, $T_0 = 273$ K the temperature at the end of the process drops below-28°C. Such strong cooling in the system may in certain cases be unacceptable. The only way to correct the situation under these initial conditions is to additionally heat the tank, i.e. to increase C_2. It is interesting to note the existence of such conditions when the temperature after decreasing it may begin to increase in the end of the process. The graphs above show examples of such processes, indicating the parameter values at which they were obtained.

The problem of lowering the temperature below the permissible limit ($-40°C$) is encountered when emptying compressed hydrogen tanks, the walls of which are made of polymeric materials sensitive to low temperatures and, at the same time, poorly conduct heat from the external environment [7].

The model proposed above makes it possible to further optimize the process by selecting a rational combination of parameters C_2, σ_0 and variation of C_2 by changing the cross-section of the gas extraction channel during the process.

4 Conclusions

The model accepted to study the emptying process of a gas tank is made in a dimensionless form. Its structure includes a block that directly takes into account the influence of thermal contact of the tank cavity with the external environment. These two factors made it possible to obtain quantitative estimation of the degree of influence of heat transfer on the process of changing the state of gas in the tank during its emptying. This makes it possible to predict the character of temperature change during the process, to determine its lower limit, to find the optimal ratio between the gas flows rate and the level of thermal insulation of the tank.

In particular, it was found that there are conditions under which the temperature of gas in the tank does not change monotonously, but according to a more complex law.
Initially, as long as the pressure and flow rate of the gas are high, the gas flows rapidly and monotonously, and when the pressure and flow rate become less than some limit, the temperature may start to rise, receiving heat from the external environment whose temperature happens to be higher than the gas temperature in the tank.

Processes of this type are shown in particular in Fig. 1 and 2. The criterion C_2 allows taking into account the specified process features already at the design stages.

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