INVESTIGATION OF GAS-CONTROLLED HEAT PIPES WITH TANKS OF CONSTANT AND VARIABLE VOLUME

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Relationships are presented for the design of gas-controlled heat pipes with tanks of constant and variable volume. The theoretical and experimental results are compared.

The prospects for using gas-controlled heat pipes (GCHP), their operating principles, different schemes for arranging the tank as regards noncondensing gas (NCG), recommendations on construction, etc., have been elucidated in detail in the literature. The majority of authors propose a mathematical model that does not take account of diffusion transfer in the condensation section, i.e., assume an ideally clear boundary between the steam–vapor mixture. Comparing the experimental and computational results verifies the legitimacy of such an assumption. However, it should be emphasized that diffusion transfer not only "spreads" the vapor front, but also contributes to the entry of the heat carrier into the tank with NCG [1-3]. Results of investigating this question are presented in [3]. A relationship obtained on the basis of the Fick law

$$\frac{c_2}{c_1} = 1 - \exp\left(-DS_C\,\tau/l_C V_b\right), \tag{1}$$

where c_1 is the steam concentration at the end of the condensation section and c_2 is the steam concentration in the tank, can be used for an approximate estimate of the quantity of heat carrier diffusing into the tank.

Diffusion transfer is apparently not the single path for steam incursion into the tank. For example, vibration is noted in [1] as a possible cause. Moreover, preliminary investigations conducted by the author of the paper permit the assumption that the heat carrier goes into the tank during the emergence of the GCHP in the stationary mode. It is noted in [1-4] that the presence of heat carrier in a tank with NCG at tank temperatures close to the steam temperature in the active zone influences the thermostatic characteristics of the GCHP substantially.* Taking this into account, it is expedient to consider the peculiarities of GCHP operation for significant temperature fluctuations in the zone of the heat fault and of the medium surrounding the tank.

In the general case, the pressure of the blocking vapor mixture can be represented as

$$P = P' + P_{\text{NCG}}\,. \tag{2}$$

Since the starting and transient characteristics of the GCHP are monotonic, the working range of the thermal parameters completely enclose the stationary modes for the extremal parameters. The relationship

$$\frac{P_{s1} - P_1'}{T_1'} V_b = \frac{P_{s2} - P_2'}{T_2'} (V_b + \Delta V_{c2}), \tag{3}$$

where the subscript 1 corresponds to the mode with maximum parameters and 2, to the mode with minimum parameters, is valid on the basis of standard assumptions for a GCHP with a constant-volume tank. Performing algebraic manipulations, we obtain

*The partial steam pressure corresponds to the temperature of the vapor mixture according to the elasticity curve (if the mixture temperature at the end of the condensation section is less than in the tank, the partial steam pressure under the effect of the concentration gradient will diminish to the saturation pressure at the end of the condensation section).

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TABLE 1. Characteristics of Experimental Specimens

	Name	Notation	Value
HEAT GUIDE	Length of evaporation section	l_e	0,2 m
	Length of transport section	l_{tr}	0,08 m
	Length of condensation section	l_{co}	0,1 m
	Shell diameter	d_{sh}	$(14\times 0{,}5)\cdot 10^{-3}$ m
	Shell material	Stainless steel	
	Diameter of capillary	d_{cap}	$(13\times 0{,}65)\cdot 10^{-3}$ m
	Material of capillary	Brass	
	Wire diameter of capillary	d_w	$0{,}1\cdot 10^{-3}$ m
	Cell dimension of capillary	a	$0{,}18\cdot 10^{-3}$ m
	Porosity of capillary	ε	0,746
	Effective heat conduction coefficient	λ_{eff}	1,0 W/m·deg
	Heat carrier	Ethyl alcohol	
Tank for NCG	Volume of rigid tank	V_b	$696\cdot 10^{-6}$ m^3
	Bellows volume	V_b	$85\cdot 10^{-6}$ m^3
	Bellows elongation factor	K	0,142 m/bar
	NCG	Air	

$$V_b = \frac{\Delta V_{c2}}{A-1}, \tag{4}$$

where

$$A = \frac{T_2'}{T_1'}\cdot\frac{P_{s1}-P_1'}{P_{s2}-P_2'}. \tag{4a}$$

Since the temperature of the vapor mixture in an unheated tank does not exceed the temperature at the end of the condensation section under real conditions, the relationship (4a) can be represented as

$$A = \frac{T_2'}{T_1'}\cdot\frac{f(T_{s1})-f(T_1')}{f(T_{s2})-f(T_2')}. \tag{4b}$$

When a heated ("hot") tank is used, the temperature T_i' will specify the maximum partial steam pressure possible for the i-th mode. This fact is analyzed in greater detail in [1].

It follows from (4) that a GCHP with a constant-volume tank is acceptable, in principle, just for A > 1. Otherwise, because of the change in partial steam pressure, the quantity of the vapor mixture (meaning the total pressure) in the tank changes inadmissibly, i.e., the saturation-temperature fluctuations in the active zone will exceed the admissible ones independently of the tank dimensions. Therefore, the complex A, due to the relationship between the intervals of the temperature changes of the vapor mixture in the tank and of the steam in the active zone, characterizes the possibility, in principle, of using a GCHP with a constant-volume tank under specific conditions.

Let us consider a GCHP with a variable-volume tank. By analogy with (3), we can write for such a system

$$\frac{P_{s1}-P_1'}{T_1'}(V_{b2}+\delta V_b) = \frac{P_{s2}-P_2'}{T_2'}(V_{b2}+\Delta V_{c2}), \tag{5}$$

where

$$V_{b2}+\delta V_b = V_{b1}. \tag{5a}$$

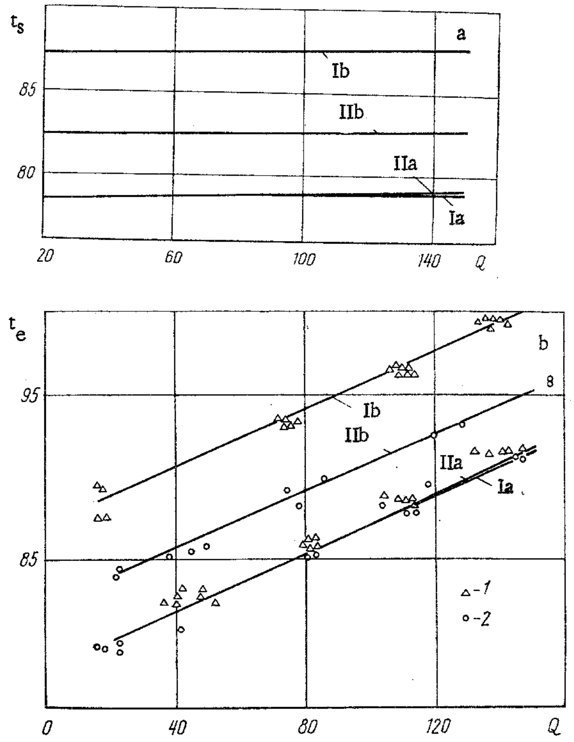

Fig. 1. Graphs of the dependence $t_s = f(Q, t', t_{h-d})$ (a) and $t_e = f(Q, t', t_{h-d})$ (b): 1) GCHP with a rigid tank; II) GCHP with a bellows; a) at $t' = 27°C$, $t_{h-d} = 27°C$; b) at $t' = 57°C$, $t_{h-d} = 42°C$; solid curve is the result of the computation; 1) experimental results for a GCHP with a rigid tank; 2) experimental results for a GCHP with a bellows. Q is measured in watts.

After manipulation,

$$V_{b2} = \frac{\Delta V_{c2} - A\cdot\delta V_b}{A-1}. \quad (6)$$

An analysis of (6) shows that the constraint characteristic for a GCHP with a constant-volume tank does not exist for a GCHP with a variable-volume tank if $A\cdot\delta V_b > \Delta V_{c2}$ † This is explained by the fact that the increment in partial steam pressure in the mixture is compensated partially by the diminution in partial pressure of the gas due to the increase in tank volume, and conversely.

To obtain a computational relationship, (6) must be represented in the initial quantities. Introducing the parameter $l_i^* = V_i/S_{st}$ and performing the manipulations, we obtain

$$A(l_{b2}^* + \delta l_b^*) - (l_{b2}^* + \Delta l_{c2}) = 0. \quad (7)$$

It should be noted that no clear steam–vapor mixture boundary exists at the condensation section of a GCHP under real conditions. Taking this remark into account in the design, the condition

$$l_{c0} > l_{c1} \quad (8)$$

must be maintained, where l_{c0} is the design length of the condensation section. Then (7) becomes

$$A(l_{b2}^* + \delta l_b^* + \Delta l_{c1}) - (l_{b2}^* + \Delta l_{c2}) = 0. \quad (9)$$

The simplest example of a variable-volume tank is a bellows. The change in bellows length depends on the pressure and elastic properties. Therefore, we can write

$$\delta l_b^* = (P_{s1} - P_{s2})\cdot S_{b.e}\frac{n}{c}\cdot\frac{S_m}{S_{st}}. \quad (10)$$

†It is easy to realize the inequality mentioned since the tank cross-sectional area will, as a rule, exceed severalfold the cross-sectional area of the steam channel.

Assuming the heat transfer conditions in the evaporation and condensation sections to be constants,† we obtain

$$\Delta l_{c} = l_{c0} - \frac{RQ}{T_s - T_{h\text{-}d}}, \tag{11}$$

$$T_s = T_e - ZQ. \tag{12}$$

Taking account of (4b) and (10)-(12), we reduce (9) to the form

$$\frac{T'_2}{T'_1} \cdot \frac{f(T_{e1} - ZQ_1) - f(T'_1)}{f(T_{e2} - ZQ_2) - f(T'_2)} \left\{ l^*_{b2} + [f(T_{e1} - ZQ_1) - f(T_{e2} - ZQ_2)] \right.$$

$$\left. \times S_{b.c}\frac{n}{c} \cdot \frac{S_m}{S_{st}} + \left(l_{c0} - \frac{RQ_1}{T_{e1} - ZQ_1 - T_{h\text{-}d1}}\right)\right\} - \left[l^*_{b2} + \left(l_{c0} - \frac{RQ_2}{T_{c2} - ZQ_2 - T_{h\text{-}d2}}\right)\right] = 0 \tag{13}$$

which can be the initial relationship for the design of a GCHP with a variable-volume tank. The function f(T), which establishes the connection between the steam pressure and temperature in the saturation line, is selected as a function of the range of temperature variation and the required accuracy.

To verify the equations proposed, computational and experimental studies of a heat-pipe specimen (heat guide) with constant- (rigid tank) and variable- (bellows) volume tanks were conducted in the Heat- and Mass-Transfer Laboratory of the Odessa Technological Institute of the Refrigeration Industry. The transmitted power (in the 20-150-W range) was varied during the studies for limiting values of the range of temperature variation of the zone of heat dumping and the vapor mixture in the tank. The heat guide and tank characteristics are presented in Table 1. The factor $K = S_{b.e} \cdot n/c$ is obtained for the bellows as a result of preliminary calibration. The following are taken as initial parameters for the study:

$$Q = 20\text{-}150\ \text{W}; \quad P_{s2} = 1\ \text{bar}; \quad t'_1 = 57\,°\text{C};$$

$$t'_2 = 27\,°\text{C}; \ t_{h\text{-}d1} = 42\,°\text{C}; \ t_{h\text{-}d2} = 27\,°\text{C}.$$

For the computations, (13) was converted to

$$\frac{T'_2}{T'} \cdot \frac{P_s - P'}{P_{s2} - P'_2} = \frac{l^*_{b2} + \left(l_{c0} - \frac{RQ_2}{T_{s2} - T_{h\text{-}d2}}\right)}{l^*_{b2} + (P_s - P_{s2})K\frac{S_m}{S_{st}} + \left(l_{c0} - \frac{RQ}{T_s - T_{h\text{-}d}}\right)} \tag{14}$$

which was solved relative to the steam temperature in the active zone of the GCHP by using a graphical interpretation of the elasticity curve. Hence, K = 0 was taken for a GCHP with a constant-volume tank. Then the shell temperature at the evaporation section was determined by means of (12).

Experimental investigations were conducted on a test stand whose diagram is given in [5]. The experimental heat guide was at a 90° angle to the horizontal line. In addition, the test stand was provided with tank heating systems and a cooling heat carrier. The temperature of the vapor mixture in the tanks was maintained to ±1.5°C accuracy and the temperature of the cooling heat carrier, to ±2.0°C. The following were measured during the experiment: the transmitted heat load Q, W; the temperature field of the heat-guide shell; the temperature of the vapor mixture in the tank t', °C; the temperature of the cooling heat carrier $t_{h\text{-}d}$, °C; and the temperature of the environment t_{en}, °C. All the measurements were conducted in the stationary mode.

The experimental data obtained completely verified the validity of the computational relationships presented above. It is important to emphasize that no special measures were taken to bring the heat carrier into the tank, except for the multiple starts, and because of diffusion a sufficient quantity of heat carrier, for example, would flow into the rigid tank during several months of continuous operation [according to (1)]. The assumption of the possibility of heat-carrier inflow into the tank during the time of starting the GCHP is thereby confirmed indirectly.

Results of the investigations are represented in the forms $t_s = f(Q, t', t_{h\text{-}d})$ and $t_e = f(Q, t', t_{h\text{-}d})$ in Fig. 1a, b. It is seen from Fig. 1a that use of a bellows whose volume is approximately one-eighth the volume of a rigid tank would, other conditions being equal, permit a significant diminution in the range of variation of t_s: for the rigid tank, 78.5-87.7°C; for the bellows, 78.5-82.7°C. Correspondingly, the range of variation of t_e would be (Fig. 1b) 80.0-100.4° and 80.0-95.4°C.

†In the general case, the heat-transfer coefficients in the evaporation and condensation sections are variables.

The agreement between the computational and experimental results permits recommendation of the equations presented for engineering methods of designing a GCHP.

If the characteristic obtained for the GCHP with a variable-volume tank is considered as the condition for a technical design, then (4a) can be used to estimate the acceptability of a GCHP with a constant tank. Since $A < 1$ in the case under consideration, the GCHP with a constant-volume tank is not, in principle, acceptable for the conditions mentioned, independently of the tank size. For such a construction (other parameters being equal), the temperature T_{S1} can be determined from the condition $A > 1$,

$$f(T_{s1}) > [f(T_{s2}) - f(T'_2)] \frac{T'_1}{T'_2} + f(T'_1). \qquad (15)$$

A computation using (15) permits the following conclusion. Independently of the size of a constant-volume tank, the saturation temperature in the active zone of the GCHP with maximum parameters will not be less than 87°C for the conditions mentioned.

Analyzing the data presented, the deduction can be made that for significant temperature fluctuations in the heat-dumping zone and the tank environment, a constraint associated with the change in partial steam pressure in the tank exists, in principle, for the thermostat characteristics of a GCHP with a constant-volume (under the condition that the heat carrier enters into the tank). Using a GCHP with a variable-volume tank permits removal of the mentioned constraint.

NOTATION

P	is the pressure;
V	is the volume;
T, t	are the temperatures;
l	is the length;
S	is the cross-sectional area;
n	is the quantity of bellow corrugations;
c	is the bellows stiffness;
Q	is the heat flux;
R	is the thermal resistivity per unit length of the condensation section;
τ	is the time;
D	is the coefficient of diffusion;
Z	is the thermal resistivity of the evaporation section.

Indices

e	is the evaporation section;
c	is the condensation section;
st	is the steam channel;
Δ	is the part of the steam channel blocked by the NCG;
s	is the saturation;
b	is the tank (bellows);
b.e	is the bellows endface;
m	is the mean (for the bellows);
NCG	is the noncondensing gas;
'	is the partial steam component in the tank;
h-d	is the heat-dumping zone;
en	is the environment;
C	is the connecting pipe (between heat guide and tank).

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DEGASSING DURING PROLONGED HEAT-PIPE OPERATION

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Processes of noncondensing-gas liberation, which affect the service life of heat pipes of the low-temperature range, are examined and analyzed. A method of computing the degassing is proposed and a comparison with available experimental results is made.

The liberation of a noncondensing gas in the inner cavity of a heat pipe was detected during service testing of low-temperature heat pipes [1, 2], where this gas, on being accumulated with the lapse of time during heat-pipe operation, will collect in the condensation zone and diminish it, thus possibly resulting in failure of the heat pipe.

The authors of [2] made an attempt to find an expression governing the mass of gas being liberated as a function of the temperature by means of the results of experimental investigations of stainless steel–water heat pipes. The small quantity of experimental points and their spread indicate the failure of these tests. An Arrhenius model [3] was used in [1] to analyze the service tests of a stainless steel–water heat pipe.

The mass flow rate of hydrogen evolution m, the time t, and the temperature are connected by the relationship

$$m(t_1T) = q(t)F(T), \tag{1}$$

where F(T) is the displacement coefficient determined from the Arrhenius equation

$$F = \mathrm{const} \cdot \exp \frac{\Delta G_0}{kT}. \tag{2}$$

Baker [1] established a temperature dependence of degassing, while Anderson et al. [4] studied stainless steel pipes; however, the results obtained have a particular character and require an experimental determination of the constant.

A complex approach to degassing processes in low-temperature heat pipes is considered in this paper, and although an analytical examination is carried out for heat pipes with heat carriers containing hydrogen, such as water, acetone, ammonia, etc., the method of computation proposed below can also be extended to other heat carriers.

Many factors affect the quantity of gas being liberated in a heat pipe; the fundamental ones under the condition of maintaining vacuum cleanliness and outgassing of the working fluids are the following: 1) thermal dissociation of the working fluid; 2) chemical dissolution of the structural material in the working fluid; 3) electrochemical dissociation of the working fluid.

The last two factors should be considered as a set, since each affects the other. It must be noted that all the above-mentioned processes will be observed to a greater or lesser degree during the operation of any heat pipe; hence, each of the processes named above yields its contribution to the total quantity of noncondensing gas being liberated in the heat pipe, i.e.,

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