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Theoretical and Experimental Investigation of a Single-Stage Positive Displacement Compressor Work Processes with a Two-Phase Working Medium

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THEORETICAL AND EXPERIMENTAL INVESTIGATION OF A SINGLE-STAGE POSITION DISPLACEMENT COMPRESSOR WORK PROCESSES WITH A TWO-PHASE WORKING MEDIUM

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THESIS

At the present time one of the main ways to increase the operation efficiency of the positive displacement compressors is to improve the cooling of the gas to be compressed.

It can be gained by injecting the cooling liquid drops into the compressed air.

The analysis carried out on the existing methods of calculation of the work processes revealed some drawbacks, such as:

1. discharge, suction and expansion processes have not bee considered at all;

2. the assumptions taken do not give a full agreement with the experimental data;

3. the work spent on the liquid injection was neglected.

To investigate the operation of a positive displacement compressor with a two-phase working medium the following mathematical model based on a number of equations is suggested:

I. <u>For A Gas Phase</u>:

a. thermodynamic equation for the variable mass on a condition of inner and outer migration of heat transfer agents;

b. mass equation;

c. mixing equation;

d. equation of state.

II. <u>For the Fluid</u>:

a. equation of a heat balance;

b. mass equation;

c. mixing equation.

III. <u>Equation of Motion for Suction and Discharge Valves Open Area Change</u>.

The elementary summary contact heat exchanger, taking into consideration the heat interaction between the phases of the working medium and the surface of a control space was defined according to the law of Newton-Rihman.

The mass heat exchange between the phases of the working medium was computed according to Maxwell law, taking into account the influence of the state variable parameters of the gas phase and the surface curvature between the phases on a diffusion coefficient.

A mathematical model developed for the positive displacement compressor with a two-phase gas-liquid system was realized in the programs called in algorithmic language as "Fortran-2" and "Fortran-4" and was applied while making calculations for piston and vane compressor.

The comparison of theoretical calculations with experimental data obtained for the piston compressor with a two-phase gas-liquid system proved that the discrepancy in defining momentum state variable parameters for the work processes is within the limits of 10%, but the definition of such main compressor characteristics as its capacity and an indicated efficiency is within the limits of 3%, which fits the developed mathematical model.

Taking the mathematical model of the work processes into account, the analysis of the effect of the injected liquid main parameters, such as quantity, dispersivity and temperature, the pressure ratio and the speed of a crankshaft rotation on the operation of a piston compressor with the added liquid was carried out.

I. INTRODUCTION

One of the most effective ways to increase an economic efficiency of the positive displacement compressors is to add a cooling liquid in a highly dispersed condition into the gas being compressed (1).

Addition of a fluid phase helps to reduce greatly a leakage as well as a mass change of the working medium and provides an intensive increase of a discharge process and improvement of the efficiency and the output of a compressor (2,3).

Nevertheless that at the present time in the USSR (1-7) as well as abroad a great experience has been gained in the field of designing and operation of the positive displacement compressors with a gas-liquid two-phase system, the theoretical problems of their work processes still need their solution.

The analysis of the theoretical methods available at present, which are used to calculate work processes of a positive displacement compressor with a gas-liquid system was carried out and brought the following calculations:

a. no calculation for discharge, suction and expansion processes were made;

b. with calculating a compression process, assumptions far from experimental results were taken (6,7). It concerns an assumption that the injected liquid evaporates either momentally or uniformly, or the system liquid-vapor during the whole period of investigation is in a saturated state;

c. work, spent on liquid injection was not taken into account. The foregoing reasons interfered with an aspect estimation of the injected liquid influence on the efficiency and the capacity of the compressor.

The given paper includes the main theoretical solutions, allowing to compute the work processes of a positive displacement compressor with a two-phase working medium, the way they fit a mathematical model by comparing with the experimental data as well as the main results of a numerical investigation of the working processes, which help to realize physical phenomena occurring in the compression chamber of a positive displacement compressor with a two-phase working medium.

II. MAIN THEORETICAL APPROACHES OF DEVELOPING A MATHEMATICAL MODEL OF THE POSITIVE DISPLACEMENT COMPRESSOR WORK PROCESSES WITH A TWO-PHASE WORKING MEDIUM

To develop a mathematical model the following main assumption should be taken into consideration: gas medium is continuous and follows the laws taken for an ideal gas; liquid drops have a spheric form; the cooling liquid drops are the same size; no crushing and coagulation of drops in a gas current; a speed of liquid drops in relation to a gaseous phase is rather slow, no discrepancy in temperature distribution along a drop radius and the heat and mass exchange processes in the compressor vanes are quasistationary.

The mathematical model of the positive displacement compressor work processes with a two-phase gas-liquid system is based on the following equations:

1. <u>For a Gas Phase</u>

   a. thermodynamic equation for a variable mass change on a condition of inner and outer migration of heat transfer agents;

   b. mass equation;

   c. mixing equation;

   d. equation of state.

2. <u>For the Fluid</u>

   a. equation of a heat balance;

   b. mass equation;

   c. mixing equation.

3. <u>Equation of Motion for Suction and Discharge Valves Open Area Change</u>

The system of equations to define the change of state variable parameters of the working medium in any control space of a compressor with a gas-liquid two-phase system is represented by:

$$dU = dQ_{\Sigma} - dL + \Sigma i_n dM_n - \Sigma i_o dM_o + \Sigma i_{\Phi n} dM_{\Phi n} - \Sigma i_{\Phi o} dM_{\Phi o} \qquad (1)$$

$$dM = \Sigma dM_n - \Sigma dM_o + \Sigma dM_{\Phi n} - \Sigma dM_{\Phi o} \qquad (2)$$

$$T = (U-g_2Mr_o)/(g+C_{vB}M+g_2C_{vn}M) \quad (3)$$

$$P = MR'_{CM}T/V \quad (4)$$

$$dT_K = [d_K\Sigma F_K(T-T_K)d\tau - r_o dM_{\Phi n,o}]/[4/3\pi_1 Z_K^3\rho_K N_K C_K] \quad (5)$$

$$T_K = [M_K T_K+(1-X_n)T_{Kn}\Sigma dM_n/X_n]/[M_K+(1-X_n)dM_n/X_n] \quad (6)$$

$$dM_K = dM_{\Phi o}-dM_{\Phi n}-[(1-X_o)/X_o]\Sigma dM_o+[(1-X_n)/X_n]\Sigma dM_n \quad (7)$$

$$Z_K = \sqrt[3]{3M_K/(4\pi\rho_K N_K)} \quad (8)$$

$$F_{BC} = f_1(\tau) \quad (9)$$

$$F_H = f_2(\tau) \quad (10)$$

Equations (9) and (10) for positive displacement compressors, having blocking elements describe their dynamics in the form:

$$m_{Kn}\frac{d^2h}{d\tau^2} = P_r - C_{nP}(h_o + h) \pm G_{Kn}\cos d_o \quad (11)$$

The elementary summary contact heat exchanger dQ_Σ includes the heat interaction between the phases of the working medium and the surface of a control space and is defined according to Newton-Rihman law.

Mass heat exchange interaction between the phases of the working medium is computed according to Maxwell law, taking into account the dependence of a diffusion coefficient from state variable parameters of a gaseous phase and the interface curvature.

Elementary contour work dL is based on the work spent on the volume change at the expense of a moving mechanism and the change of the phase mass concentration of a condition of outer migration of heat transfer agents.

Enthalpy and inner energy of a gaseous phase for the working medium is defined by summing up enthalpies and inner energies of vapor and air.

Thermophysical properties of the working medium phases and their components over the temperature range being investigated are interpolated by polynoms to the 3-d and 4-th power. Mixing laws are used to define thermophysical properties of the gaseous phase of the working medium.

A numeric analysis of formulas used to calculate the average liquid drop radius was carried out and showed, that the most exact definition of the total heat transfer surface of the cooling liquid and its mass is obtained from the formulas (8).

The working medium discharge when it is flowing through gas-distribution valves was computed from the formula, suggested by Academicians S.S. Cutateladze and M.A. Styrikovich (8), which for the case of a gas phase discharge may be expressed:

$$dM_o = \eta_{Kn} X_o \rho_{CM} F_{Kn} \sqrt{\frac{2}{C_1}[(P-P_o)+\frac{K}{K-1}(P^{\frac{K-1}{K}} - P_o^{\frac{K-1}{K}})B_1^{\frac{1}{K}}} \, d\tau \tag{12}$$

The analysis of the power consumption by a compressor revealed that for a positive displacement compressor with a two-phase gas-liquid system the power used for a liquid injection is highly important.

Hence, the power consumed by a compressor may be represented:

$$N_{KoM} = N_{UHg} + N_{TP} + N_{BnP} + \Delta N_{neP} \tag{13}$$

The comparison of the economical operation of different types of injectors led to the conclusion, that one of the simplest in design and the most effective in operation is a centrifugal pressure-jet burner.

A calculation of injectors was carried out by incorporating both the theory worked out by G.N.

Abramovich and the experimental data obtained by A.G. Bloh and E.S. Kichkina (8).

The mathematical model built on the basis of the foregoing theoretical statements the positive displacement compressors work processes was realized in the programs called in algorithmic language as "Fortan-2" and "Fortan-4" and was applied when computing piston and vane compressors with a two-phase working medium.

III. EXPERIMENTAL INSTALLATION AND THE CHECK-OUT OF ADEQUACY OF THE MATHEMATICAL MODEL

To obtain the information required to develop a mathematical model and to check up its adequacy, an experimental installation with a piston compressor and a cooling liquid injection was made.

For experimental investigation a momentum pressure in the compression chamber and under piston cavity was measured by means of a foil resistance strain gauge; a momentum temperature of a gaseous phase of the working medium was measured with a special device during 1-3 cycles; the temperature of the inlet air, injected liquid and the cylinder surface (along the cylinder sleeve perimeter and along a generating line) was measured with the help of chromium-nickel alumel thermocouples; the humidity of the injected and forced air, the amount of the injected liquid, the compressed air discharge and the power consumed by the compressor were measured as well.

The average radius for the liquid drops to be injected into the compressor was calculated on the basis of the received experimental data.

A possible error in measuring the above-mentioned values was defined.

By applying the method of a regression analysis to the experimental data we gain a formula to determine the average temperature of the compression chamber surface in a piston compressor of low capacity:

$$T_{Cm} = 277.2 + 0.564(E_{Cm}-1)-1.01d_{BnP} + 1.034(T_{KB}-273)+0.011n_o{}^{\sigma} \qquad (14)$$

and it was found that the average surface temperature of the compression chamber is highly dependent from the injected liquid temperature; furthermore, it depends on the relative anorent of the injected liquid kg/kg, on the pressure ratio and is slightly affected by the rotation speed of a crankshaft.

Comparison of a theoretical investigation with the experimental data, where different values of the injected liquid, its dispersity, temperature, the rotation speed of a crankshaft and the pressure ratio were involved, showed that the discrepancy in defining momentum state variable parameters for different work processes is with the limit of 10%, but for the main external characteristics of a compressor (capacity and efficiency) is within the limits of 3% which fits the mathematical model adequately well.

Figure 1 illustrates the results of comparison of different indicating diagrams received theoretically and experimentally when investigating work processes of the compressor under the following conditions:

$E_{Cm} = 6$; $n_o\sigma = 444$zev/min; $T_{BC} = 340K$; $T_{KBC} = 331.5K$;

$X_{BC} = 0.1712$; $Z_{KBC} = 113$mkm; $g_1 = 0.00912$; $g_2=0.9908$;

$a_M = 2.875\%$.

The given results help to make the following conclusions:

1. A good agreement of the theory with the experiment for a momentum pressure in the compression chamber leads to the conclusion, that the mass and heat exchanges taking place in a compressor cylinder are accurately enough described by the mathematical method.

2. A discrepancy in defining a momentum pressure in suction and discharge processes may be accounted for taking the following assumptions, such as: there was no pressure change in compressor cavities and pipelines a fluid film on the saddle surface and on the surface of the pressure rise limiters does not influence the motion of the blocking elements, besides, there were some errors in experimental investigation.

   However, despite the discrepancy exists maximum pressure losses in inlet valves and in suction valves are defined by the model accurately enough, which justifies the assumptions taken for calculation of the working medium discharge when it flowing through gas-distribution valves.

IV. THE RESULTS OF INVESTIGATION

Inclusion of a liquid phase into the working medium helps to reduce the temperature of a gas phase in the compressor cylinder over the whole range of its operation.

The highest reduction of the compressed gas temperature is observed at the end of the process of compression.

So, we find out when the relative amount of the injected liquid comes to 0.43 kg/kg, its temperature is 278 K and the drop radium is 50 mm, the reduction of the temperature at the end of compression with the pressure ratio of 5 comes to 50 K (Fig. 2).

With the increase of the amount of the injected liquid the quantity of heat Q_K brought to the cooling liquid drops, which comprises a part of the total heat quantity absorbed in the process of compression grows, which results in the temperature reduce at the end of compression.

At X_{bc}=0.8, T_{kbc}=288K, r_{kbc}=50mkm, the relationship between Q_K and Q_Σ comes to 56%, but at X_{bc}=0.5 it is equal to 84%.

The way of temperature change in the liquid phase of the working medium in the compressor cylinder is similar to the way of temperature change of the gaseous phase of a cylinder, however, the temperature of phase is essentially different. The increase of the injected liquid value as a result of density increase of the working medium causes a rise of pressure losses in the processes of suction and discharge, but an intensive interphase heat exchange result in pressure reduction during compression and pressure increase during expansion.

A reduction of drops size of the injected liquid scales up the surface of interphase heat exchange and the coefficient of convective heat exchange which causes a temperature drop of a gaseous phase in work processes of a compressor and a rise of a relationship between Q_K and Q_Σ.

The increase of the intensity of interphase heat exchange results in a temperature rise of the cooling liquid drops and intensity of a mass exchange.

At Z_{KBC} = 20mkm; d_{Bnp} = 0.05kg/kg; T_{KBC} = 278K the relationship between heat value, taken from the cooling liquid drops at the expense of evaporation and the heat value brought to the cooling liquid drops come to 36%, but at Z_{KBC} = 10mkm it comes to 76% (Fig. 3).

So, we may conclude, that when we inject liquid drops with Z_{KBC} < 20mkm a greater amount of heat brought to the drops is taken away because of evaporation.

When liquid drops are added with $Z_{KBC} \geqslant$ 40mkm, T_{KBC} = 278K, g_{2BC} = 0.01, d_{BnP} = 0.1kg/gk.

it can be observed that in the compressor cylinder during the process of compression a condensation of vapor, contained in the compressed air occurs on the drops instead of evaporation of the cooling liquid drops.

The analysis of the injected liquid main parameters influence on the components of the power consumed by a compressor revealed, that their influence is rather complicated and contradictory. So, the increase of the amount of the injected liquid brings the reduction of the specific values of the rated power, of the power spent on compression and vapor transfer as a result of a less intensification of a mass exchange process due to a temperature drop of the liquid phase of the working medium, but increases power losses in valves and power expenses for injection.

The investigation carried out stated, that the increase of the compressor capacity can be gained only at the expense of an intensive cooling of the absorbed gas by injecting it into the suction conduit and reducing the value of a dead space, because the liquid addition increases the expansion process, reduces the density of the absorbed gas as a result of pressure losses rise in an inlet valve. decreases the amount of the inlet gas as a result of inclusion of into a compressed air of the cooling liquid drops and their evaporation, which finally reduces a compressor capacity.

The effect of a liquid injection increases with a pressure ratio rise. It helps to raise the pressure ratio in a compressor stage (Fig. 4) and to obtain a maximum efficiency and a safe operation of a compressor.

NOMENCLATURE

u - total inner energy

T - temperature of a gas stage

M	- mass of a gas phase
V	- volume of a gas phase
P	- pressure of the working medium
M_K	- mass of a liquid phase
T_K	- temperature of a liquid phase
N_K	- number of drops
z_K	average radius of a drop
c_{vB}	- specific isochoric heat capacity of air
c_{vn}	- specific isochoric heat capacity of water steam
r_o	- heat of steam formation
R'_{CM}	- gas constant for a gas phase in a working medium
g_1, g_2	- mass concentrations of steam and air in a gas phase of a working medium
X	- mass concentration of a gas phase a working medium
P_r	- gas force
d_K	convective heat exchange coefficient
ΣF_K	- surface of heat exchange between a compressed air and liquid drops
ρ_K	- density of the liquid
c_K	- specific heat capacity of the liquid
M_{Kn}	- mass of a moving element in a gas-distribution valve
G_{Kn}	- weight of a moving element in a gas-distribution valve
c_{np}	- spring rigidity
h	- running elevation height
h_o	- value of a preliminary tension of a moving element
τ	- time

d_o	- angle formed between the direction of a moving element and a vertical line
dQ_Σ	- elementary summary contact heat exchange
dL	- elementary contour work
η_{Kn}	- coefficient of discharge
ρ_{CM}	- density of the working medium
F_{Kn}	- area of an open cross-section
K	- adiabat indicator of a gas phase
P_o	- pressure in a control space, where the working medium flows
ρ'_{oCM}	- gas phase density in a control space where the working medium flows
N_{KOM}	- power consumed by a compressor
N_{UHG}	- indicated power
N_{TP}	- power used to overcome friction
N_{BnP}	- power used for injection
ΔN_{neP}	- power lost in energy transmission from the energy source to the compressor
T_{Cm}	- average temperature of the compression chamber surface in a piston compressor
E_{Cm}	- pressure ratio
$n_{o\sigma}$	- number of revolutions per minute
T_{BC}	- temperature of suction
T_{KBC}	- injected drops temperature in the process of suction
B_1, C_1	- constants
Z_{KB_C}	- average radius of drops for suction
$g_1 B_c$	- mass steam concentration during suction
$g_2 B_C$	- mass air concentration during suction

a_M - relative value of a dead space

X_{BC} - relative gas phase concentration during suction

Q_K - heat added to the cooling liquid drops

Q_Σ - total heat value absorbed in the process of compression

d_{BnP} - relative quantity of an injected liquid

Q_{NC} - amount of heat used for liquid drops evaporation

INDICES

n - attachable element of a gas phase in a working medium at outer migration

O - separated element of a gas phase in a working medium at outer migration

Φn - attachable element of a gas phase in a working medium at inner migration

ΦO - separated element of a gas phase in a working medium at inner migration (evaporation, condensation)

BC - refers to suction parameters

FIGURES DESCRIPTION

Figure 1. Comparison of indicator diagrams obtained both experimentally and theoretically for the experimental piston compressor with a liquid injection.

Figure 2. Dependences of the temperatures change at the end of a compression process from the pressure ratio with a different amount and dispersivity of the injected liquid.

Figure 3. Dependences of a relationship change between Q_{UC} and Q_K from dispersivity and amount of the injected liquid.

Figure 4. Dependences of an indicated isothermal efficiency change from the pressure ratio with a different amount and dispersivity of the injected liquid.

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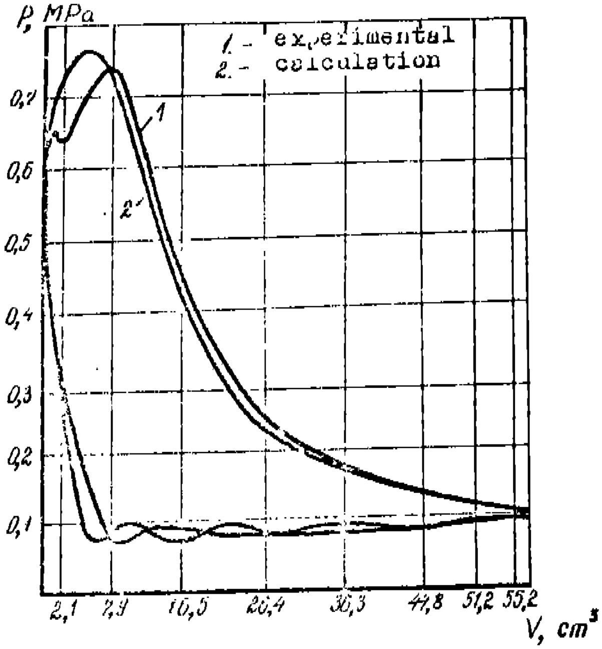
P, MPa
1 - experimental
2 - calculation
0,7
0,6
0,5
0,4
0,3
0,2
0,1
1
2
2,1
2,9
10,5
20,4
30,3
41,8
51,2
55,2
V, cm³

Fig. 1

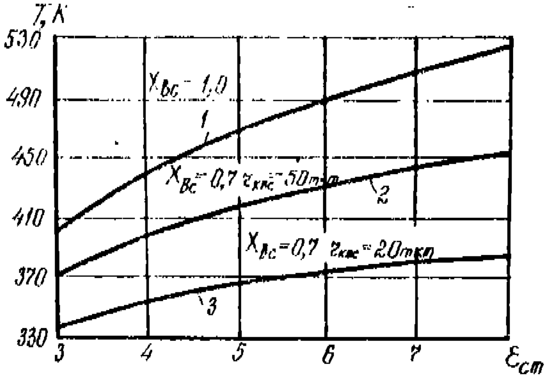
T, K
530
490
450
410
370
330
X_{Bc} = 1,0
1
X_{Bc} = 0,7 $z_{кис}$ = 50mkm
2
X_{Bc} = 0,7 $z_{кис}$ = 20mkm
3
3
4
5
6
7
ε_{cm}

Fig. 2

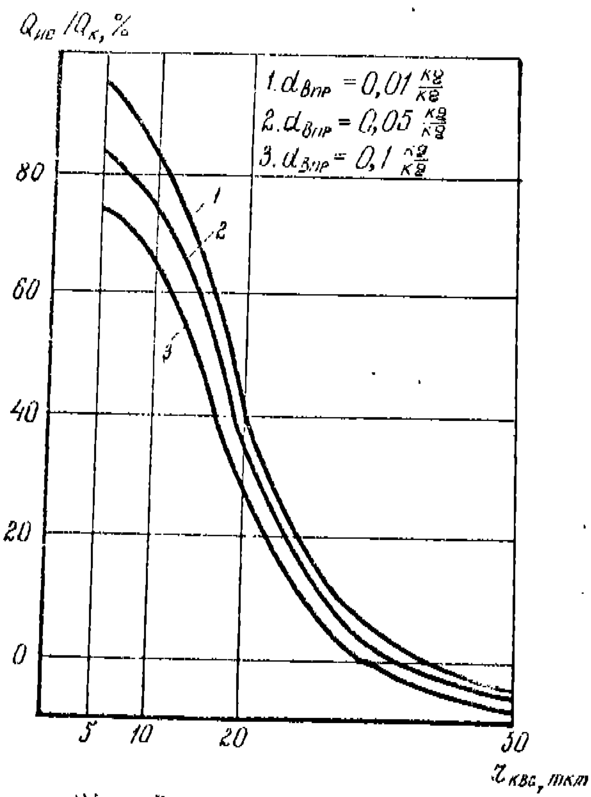
$Q_{ис}/Q_к$, %
1. $d_{Впр}$ = 0,01 кг/кг
2. $d_{Впр}$ = 0,05 кг/кг
3. $d_{Впр}$ = 0,1 кг/кг
80
60
40
20
0
5
10
20
50
$r_{квс}$, мкм
1
2
3

Fig. 3

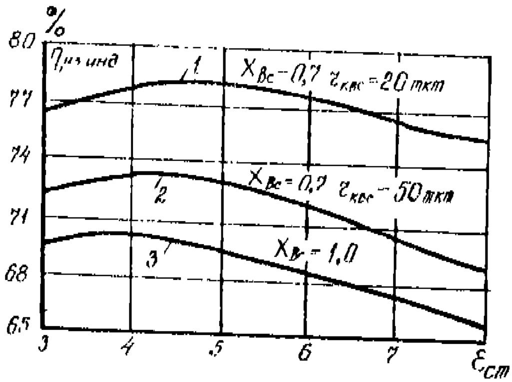
%
80
77
74
71
68
65
3
4
5
6
7
$\varepsilon_{ст}$
1
2
3
$X_{Вс}$ = 0,7 $r_{квс}$ = 20 мкм
$X_{Вс}$ = 0,2 $r_{квс}$ = 50 мкм
$X_В$ = 1,0

Fig. 4