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Research Article

TEMPERATURE OF THE COMPRESSOR SUCTION AIR AND ITS INFLUENCE ON THE EFFICIENT OPERATION OF THE COMPRESSOR UNIT

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ABSTRACT

The climatic conditions of Uzbekistan are characterized by relatively long summers with air temperature of 40-45 oC and higher, which is the reason of significant heating of air suctioned by compressor. Increase in temperature and corresponding decrease in air density contributes to 8-10% decrease in compressor unit productivity, leading to significant economic losses in the process of compressed air production.

By means of artificial cooling of suction air at the inlet to the compressor unit it is possible to increase the energy efficiency of operation of the mine reciprocating compressor unit.

The effectiveness of artificial air temperature reduction depends on the amount of temperature reduction and the type of plant drive, as well as on the purpose - to reduce the specific electric energy consumption for compression or to increase the capacity of the reciprocating compressor.

KEYWORDS

Compressor, air temperature, energy efficiency, air suction, air heating, capacity, compression, air pressure, artificial cooling.

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INTRODUCTION

The performance of a real compressor differs from an ideal compressor due to the presence of harmful volume, hydraulic resistances at the suction, leakages in the working cavity, relative humidity of the suction air and suction air heating [1].

One of the main factors affecting the efficient operation of mine compressors is air heating during suction, resulting from the resistance of the suction path and heat exchange with heated components of the unit. Under normal atmospheric conditions, the decrease in mass capacity and volumetric delivery of the equipment is due to a decrease in the density of heated air [2].

The delivery coefficient, which takes into account the influence of all factors on the performance of the actual compressor, is determined by the following formula [3]:

$$\lambda = \lambda_{vc}\lambda_{thc}\lambda_{hc}\lambda_{dc}\lambda_{mc}, \quad (1)$$

where: λ_{vc} is the volumetric coefficient;

λ_{thc} - throttling coefficient;

λ_{hc} - heating coefficient;

λ_{dc} - density coefficient;

$\lambda_{вл}$ - moisture coefficient.

There are several reasons for the warming of the air observed during suction. The air entering the compressor unit is heated by contact with the cavity walls during the suction process. The energy expended in pushing the suction air through hydraulic resistances, which is converted into heat, results in a temperature increase. In addition, the change in air temperature is caused by the mixing of the air newly entering the cylinder with the air remaining in the harmful space.

Taking into account the above, it is revealed that the decrease in the compressor unit performance is affected by the suction gas heating due to heat exchange with the surfaces of hot cylinder parts ΔT1, and the increase in the suction gas temperature due to the gas remaining in the dead space ΔT2.

Thus, at the end of the suction process, the conditional value of the air temperature in the working cavity is determined by the formula [4]:

$$T_1' = T_1 + \Delta T_1 + \Delta T_2, \quad (2)$$

where T_1 is the atmospheric air temperature, °C.

The influence of intake air heating on the plant performance is estimated by its temperature coefficient determined by the formula [4]:

$$\lambda_т \cong \frac{T_1}{T_1'} \quad (3)$$

As can be observed from the formulas, an increase in the atmospheric air temperature T1 lead to an increase in the temperature T1', which reduces the performance of the piston compressor.

Until now, the value of $\lambda_т$ could not be determined precisely because of the difficulty in determining the instantaneous temperatures at the beginning of suction [5].

The compressor capacity Q_pr taking into account the effect of temperature can be determined by the following expression [5]:

$$Q_{пр} = V_р' \cdot \frac{P_1'}{P_1} \cdot \frac{T_1}{T_1'}, \quad {м^3}/{h} \quad (4)$$

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where V_p'' is the volume of atmospheric air intake, m3/h;

T_1 and P_1 — temperature and pressure of air at the suction, respectively;

T_1' and P_1' - respectively temperature and pressure of air in the cylinder during suction.

The effect of suction temperature T_1 on the amount of work expended by the compressor to compress air can be determined from equation (5).

Total energy consumption for compression of 1 kg of air [6]:

$$l_{п.э} = \frac{k}{k-1} \cdot R \cdot T_1 \cdot \left[\left(\frac{P_н}{P_1} \right)^{\frac{k-1}{k}} - 1 \right], \qquad (5)$$

Where k-is the adiabatic index of air;

R- is the gas constant, j/(kg· deg);

P_1- pressure of inlet air, MPa;

$P_н$- pressure of injected air, MPa.

As can be seen from equation (5), in the compressor at constant relative pressure rise rise $\varepsilon = \frac{P_н}{P_1}$,,,, the compressor work for compressing and moving 1 kg of air increases with increasing temperature T1. Work spent on compression of 1 kg of air, for each 10C increase in temperature at suction T1, increases by 0.16% [7].

The climatic conditions of Uzbekistan are characterized by relatively long summers with air temperatures of 40-45 0C and higher, which is the reason for significant heating of air suctioned by the compressor. The increase in temperature and the corresponding decrease in air density contributes to an 8-10% decrease in compressor unit productivity, resulting in significant economic losses in the compressed air production process.

By artificially cooling the intake air at the inlet to the compressor unit, the energy efficiency of a mine reciprocating compressor unit can be increased.

The effectiveness of artificial air temperature reduction depends on the magnitude of the temperature reduction and the type of drive of the unit, as well as on the goal of reducing the specific electrical energy consumption for compression or increasing the capacity of the reciprocating compressor.

Undoubtedly, reducing the temperature of the compressor suction air leads to an increase in its weight capacity and increases the power of the drive. In this regard, by artificially cooling the suction air, it is possible to increase the annual compressor capacity in summer, which is especially important in the conditions of plant operation in areas with hot climate, to eliminate the need to install additional compressors or avoid replacing existing units with more powerful ones, which will give a significant economic effect [8].

Reduction of reciprocating compressor performance in summer period is primarily due to the increase in ambient temperature. As a consequence, the temperature of the air sucked in by the compressor increases, and the air density changes accordingly. Air density is commonly understood as the amount of air contained in 1 m3, its value can be determined by the following expression:

$$\gamma_{воз} = \frac{M}{V}, \quad кgs/_{м^3}, \qquad (6)$$

where M-weight of air, kgs;

V - volume of air, m3.

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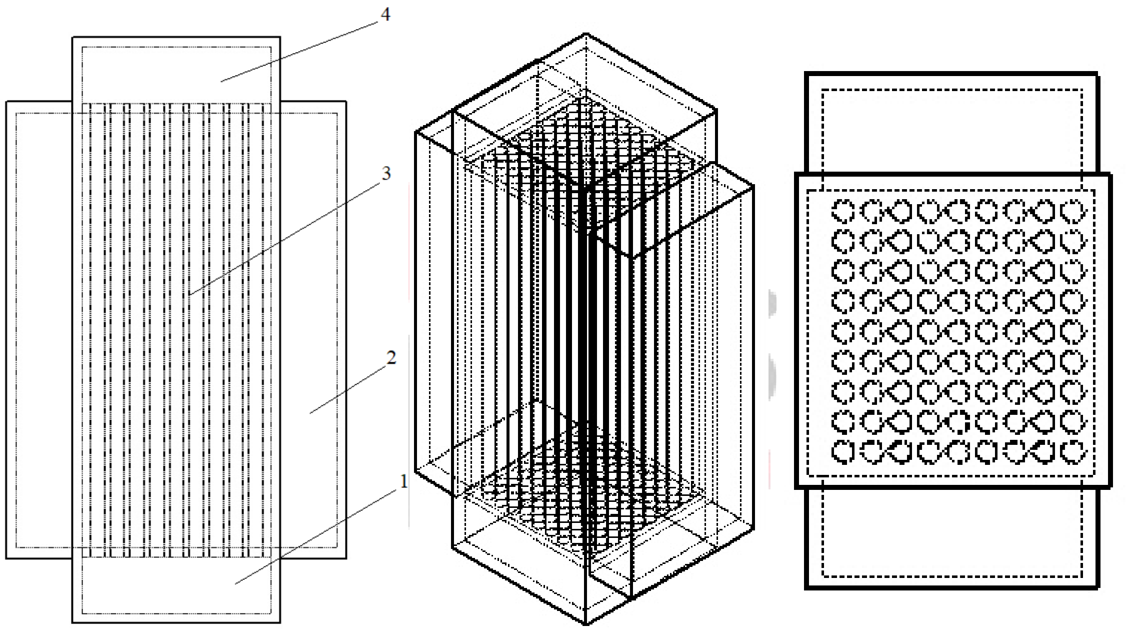
At normal atmospheric pressure (P_{nor}=760 mm.Hg) and temperature t_{nor}=15-25 °C 1 m3 air density determined by formula (6) is γ_a =1,225 kgs/(m3)

It has been established that in winter period at twin=0 °C the air density is γ_a is=1,29 kgs/m3 , while in summer period

Ts=40°C, γ_(air.l.)=1.13 kgs/m3. Hence, the air density in summer period is less than that in winter period. Thus, to ensure greater performance of reciprocating compressor units there is a need for artificial cooling of the air entering the compressor.

The temperature of the compressor suction air can be reduced by simple and cheap heat exchangers using cold water from the cooling tower, in this case it is recommended to install the cooler of the suction air between the filter and the first stage of the compressor. The proposed design of the cooler is shown in Fig. 1.



a) b) c)

1 - cooling water inlet side; 2 - housing; 3 - tubes; 4 - cooling water outlet side;

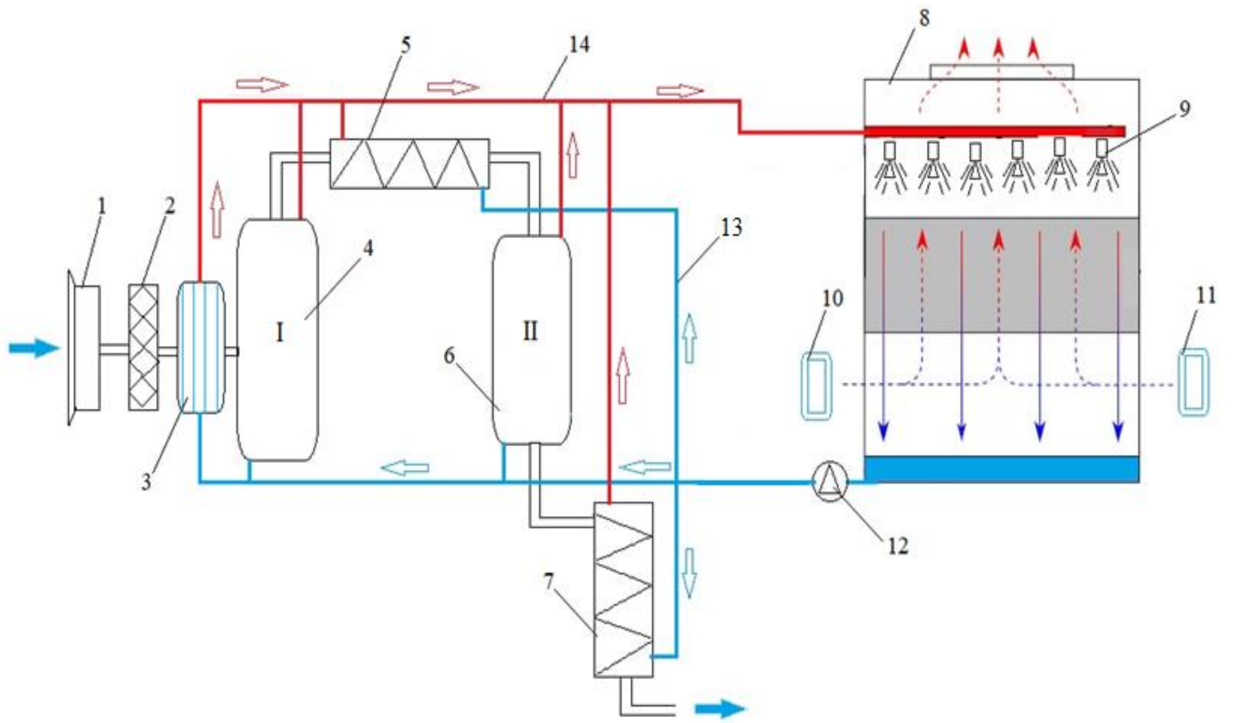
a) front view; b) general view; c) top view
b)

Fig.1. Compressor suction air cooler

Installation of coolers at the suction before the compressor creates additional hydraulic resistance to the movement of suction air, which in turn leads to an increase in energy costs of the drive unit.

In our proposed cooler design, the distance between the tubes through which the cooling water circulates is chosen in such a way that the created hydraulic resistance will not be significant.

Fig. 2 shows a schematic of an open-circuit cooling system for a two-stage reciprocating compressor using a heat exchanger to cool the suction air.



1 - air intake; 2 - filter; 3 - air cooler before the compressor; 4 - first compressor stage; 5 - intermediate cooler; 6 - second compressor stage; 7 - end cooler; 8 - cooling tower; 9 - water atomizer; 10 and 11 - fans; 12 - pump; 13 - cooled water pipeline; 14 - heated water pipeline.

Fig.2. Scheme of open-circuit cooling system of two-stage reciprocating compressor with heat exchanger for artificial cooling of suction air

Pre-cooling of the compressor intake air, shown in Fig. 2, is carried out as follows: air cooler 3 is installed between the filter 2 and the first stage of the compressor 4. Pump 12 supplies cold cooling water from the cooling tower 8 to the compressor to cool its parts, to the intermediate 5 and end coolers 7 to cool the compressed air. At the same time, the water is supplied to the air cooler 3. The water heated as a result of air cooling is fed to the cooling tower for cooling and then back to the air cooler upstream of the compressor unit.

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