Device for Researching the Cooling Intensity of Flowing Media

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Abstract— *The article describes the construction of a device for researching the effect of air flow speed on the intensity of cooling of the medium that flows in the ribbed tubes of the cooler. The measure of cooling intensity is the overall heat transfer coefficient k. It is a model of a tubular heat exchanger in which horizontal and vertical movement of the individual tubes of the heat exchange surface is possible. The device will be used to obtain information about the influence of the arrangement of the heat exchange surface on the cooling performance of the exchanger. The measured values of the relevant quantities will be used as input data for the ANSYS CFX software. The software will allow to simulate the current ratios in the radiator tubes, the temperature fields along the length of the tubes and in the individual tubes of the exchanger arranged according to its height.*

***Keywords*— Ribbed heat exchange surface, flow distributor, cooling capacity.**

I. INTRODUCTION

To achieve suitable transport conditions, tubular heat exchangers with a ribbed external heat exchange surface are used at compressor stations for cooling natural gas [1-4]. The transported natural gas is compressed to the required transport pressure at the compressor station, which causes an increase in its temperature. Since the increase in gas temperature causes an increase in the volume of transport gas, the quality of cooling becomes one of the transport priorities. By reducing the temperature, the mechanical stress on the gas pipeline pipes caused by the expansion of their material is also reduced, which positively affects the service life of the gas pipeline.

The device described in the article is intended to provide information on the possibility of intensifying the heat exchange when cooling natural gas during the operation of the cooler. Natural gas coolers have different cooling surfaces, which are characterized by different tube diameters, different tube wall thicknesses, different rib diameters, and different rib spacing. However, the most important thing is the way the tubes are arranged in the cooler. The pipes are placed in rows above each other in the so-called "alternate arrangement" (Fig. 1). The number of rows of tubes is different. Structurally, the individual rows create an arrangement of either crossed flow (Fig. 2) or current doubly crossed (Fig. 3).

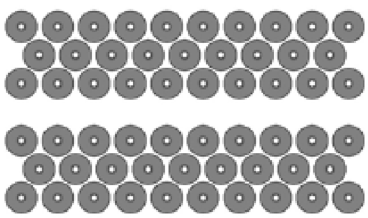

FIGURE 1: Pipe arrangement

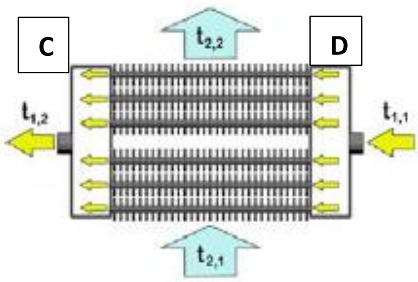

FIGURE 2: Cross current

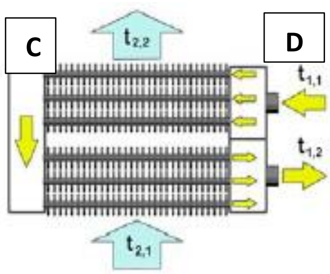

FIGURE 3: Double crossed current

In cross-flow the gas is fed to the distributor (D), from where it flows through all rows of cooling tubes in one direction. The tubes open into the collector (C), from where the gas is taken out of the cooler through the outlet pipes. With double cross current, the gas inlet and outlet to the cooler are on the same side of the cooler. In both cases, air is used to cool the gas, which is blown into the cooler by means of fans from the bottom up. Fan start-up is regulated depending on the desired output temperature.

II. DESCRIPTION OF THE PROPOSED DEVICE

In laboratory conditions, water will be used instead of natural gas (safety point of view). The designed measuring device forms a closed circuit, which consists of a tubular heat exchanger (1) - a cooler, a fan placed under the cooler (2), a circulation pump (3), an accumulation flow water heater (4), a distributor (5), a collector (6), collecting tank (7) and connecting hoses (Fig. 4). The heat exchange surface is represented by the same pipework that is used to cool natural gas at the compressor station.

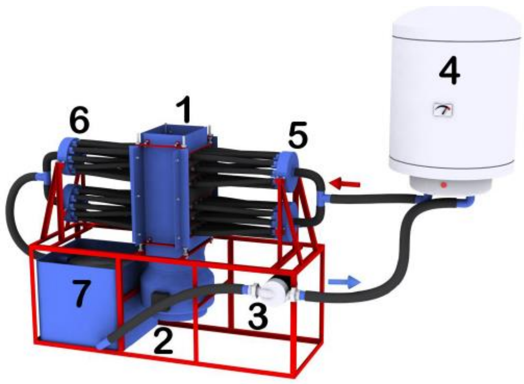

FIGURE 4: Assembly of measuring equipment

The main component of the measuring device is the tube exchanger, which is designed to ensure the greatest possible flexibility of the tube arrangement. The number of tubes placed one above the other can be up to 6, just like on a real natural gas cooler. The exchanger consists of ribbed pipes with a length of 2 m. The basic dimensions of the tube are shown in fig. 5. The individual pipes are fixed in the cooler structure in a staggered arrangement (Fig. 6). The device is currently prepared for measuring on tubes with the following dimensions: tube inner diameter d_1 = 25 mm, tube outer diameter d_2 = 30 mm, rib diameter d_r = 58 mm, rib thickness h_r = 0.65 mm, rib spacing b = 2.5 mm.

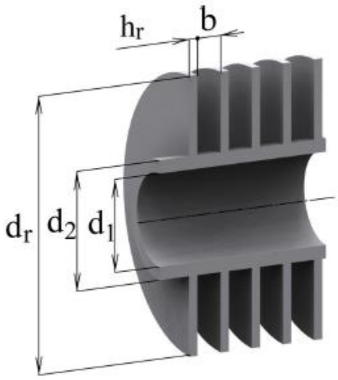

FIGURE 5: Basic dimensions of a pipe

FIGURE 6: Fixation of pipes in the structure of the device

The ends of the tubes are clamped in the measuring device between two steel bands, which are tightened using M 12 nuts threaded on threaded rods. The height adjustment of the pipes is ensured by the nuts on the underside of the steel belt. After setting the spacing between the tubes, the ends of the tubes are pressed against the steel strip using a nut. This will arrest the tubes. The tube block is then sheathed on the outside. This creates a closed space for the cooling air supply (Fig. 7).

On the sides, the exchanger casing consists of sheet metal parts with the possibility of precise adjustment of the gap between the outer tube and the shell. From the sides of the water inlet to the exchanger, the space is sealed with rubber seals (Fig. 8).

The sheathed heat exchanger is mounted on an L-shaped profile steel support frame. A VKMS 315 fan with a maximum air flow rate of 1,880 $m^3 \cdot h^{-1}$ is placed under the heat exchanger. Water entering and exiting the heat exchanger is ensured by a flow distributor - position 5 in Fig. 4. Distributors ensure uniform distribution of the flow to all pipes of the exchanger. The connection of the distributors depends on the way the water flows through the exchanger. In the case of crossed current, two distributors and 2 collectors connected according to Fig. 9. In case of doubly crossed current, one distributor and one collector will be used (Fig. 10). Water is provided in a closed circuit by a pump HGPA 25-8.0 U 180 with a maximum flow of 12 $m^3 \cdot h^{-1}$ with three levels of regulation.

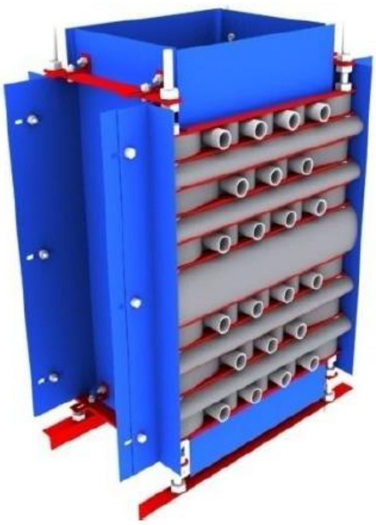

FIGURE 7: Exchanger with sheathing

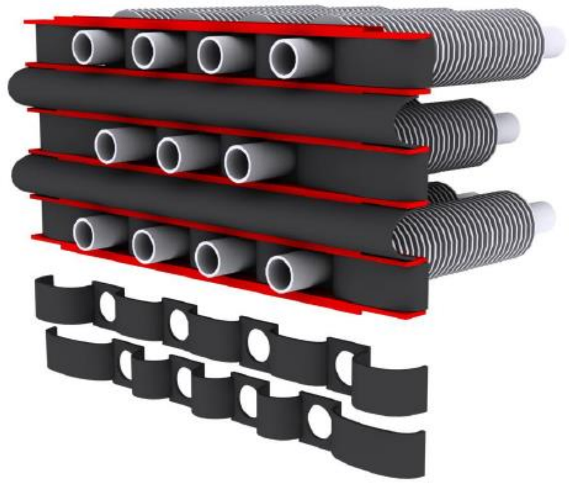

FIGURE 8: Rubber seals on the inlet side of the exchanger

FIGURE 9: Diagram of connection of distributors for cross current

FIGURE 10: Wiring diagram of distributors for double crossed current

III. BASICS OF COOLING PERFORMANCE CALCULATION

The cooling performance of the cooler generally depends on several parameters. This is primarily about the type of cooled and cooling media and their thermal-physical properties, flow rates of both media, flow method through the cooler (convex, counter-flow, cross-flow), shape, size and material of the heat exchange surface, arrangement of pipes in the cooler, etc.

On the flowing water side, the basic relationship for the cooling power P can be given in the form:

$$P = \eta \cdot Q_{m,1} \cdot (i_{1,1} - i_{1,2}) = \eta \cdot Q_{m,1} \cdot c_{p,1} \cdot (t_{1,1} - t_{1,2}) = \eta \cdot \rho \cdot Q_{V,1} \cdot c_{p,1} \cdot \Delta t_1 \text{ (W)} \quad (1)$$

where $i_{1,1}$ is the specific enthalpy of water at the inlet to the exchanger ($J \cdot kg^{-1} \cdot K^{-1}$),

$i_{1,2}$ – specific enthalpy of water at the outlet of the exchanger ($J \cdot kg^{-1} \cdot K^{-1}$),

t – water temperature (°C); the indices are the same as for the designation of the specific enthalpy of water,

$c_{p,1}$ – specific heat capacity of water ($J \cdot kg^{-1} \cdot K^{-1}$),

$Q_{m,1}$ – mass flow of water ($kg \cdot s^{-1}$),

η – coefficient respecting heat losses in the exchanger structure (1).

The total cooling performance of the exchanger can also be expressed from the heat balance:

$$P = k \cdot S_{\Sigma} \cdot \overline{\Delta t} \text{ (W)} \tag{2}$$

where k is the heat transfer coefficient referred to the outer surface area ($W \cdot m^{-2} \cdot K^{-1}$),

S_{Σ} – heat exchange surface on the outside (m^2),

$\overline{\Delta t}$ – average temperature drop (K); in the case of cross-flow of heat-carrying media, it is determined as the mean logarithmic difference in the counter current exchanger, multiplied by the coefficient ψ. Its determination is stated in [5].

With known values of physical quantities, obtained from measurements on experimental equipment, the overall heat transfer coefficient k can be determined from the equation (3)

$$k = \frac{P}{S_{\Sigma} \cdot \overline{\Delta t}} = \frac{\eta \cdot \rho \cdot Q_{V,1} \cdot c_{p,1} \cdot \Delta t_1}{S_{\Sigma} \cdot \overline{\Delta t}} \ (\mathrm{W \cdot m^{-2} \cdot K^{-1}}) \tag{3}$$

The relationship (3) will allow obtaining the average value of the overall heat transfer coefficient k for various shaped heat exchange surfaces that will be placed in the designed experimental stand. To achieve a uniform distribution of the cooled medium in all tubes of the heat exchange surface, a "distributor" was designed (Fig. 11). The uniformity of the flow distribution was tested numerically, and the distributor was solved in the ANSYS CFX simulation tool (Fig. 12).

FIGURE 11: Cross-section of the flow distributor

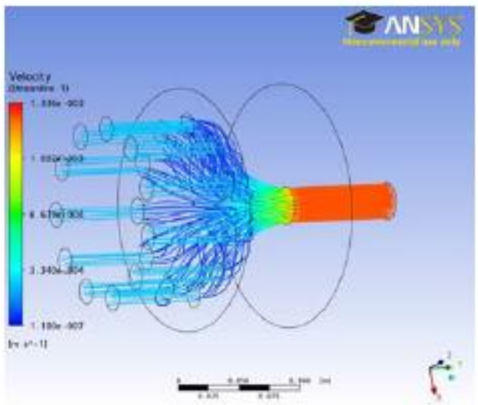

FIGURE 12: Simulation of water flow through the distributor

IV. DISCUSSION

When modeling the cooling performance, it is necessary to maintain the physical similarity between the model and the part. The heat dissipation to the surrounding air must be identical on the part and the model. In the model, water is used as the cooled liquid, in the real cooler it is compressed natural gas. It is necessary to ensure that the value of the heat transfer coefficient α from the cooled medium to the inner surface of the tube has the same value on the part and the model. This results from the equality of the Nusselt criteria for the laboratory model and the real natural gas cooler. The flow of natural gas on a real cooler is turbulent. The heat transfer coefficient in the pipe is described by the criterion equation:

$$Nu = C \cdot Re^{0.8} \cdot Pr^{0.4}$$

where C is a coefficient (for gases, C is equal to 0.021; for liquids 0.023),

Re – Reynolds criterion (1),

Nu – Nusselt's criterion (1).

The same values of the factor α on the part and the model will be achieved if the derived dependence in the form applies between the water flow rate w_w and the natural gas rate w_g:

$$w_{\mathrm{w}} = w_{\mathrm{g}} \cdot \left[\frac{0.021}{0.023} \cdot \left(\frac{\nu_{\mathrm{w}}}{\nu_{\mathrm{g}}}\right)^{0.8} \cdot \left(\frac{Pr_{\mathrm{g}}}{Pr_{\mathrm{w}}}\right)^{0.4} \cdot \frac{\lambda_{\mathrm{g}}}{\lambda_{\mathrm{w}}}\right]^{1/0.8}$$

where ν is the kinematic viscosity ($m^2 \cdot s^1$),

λ – coefficient of thermal conductivity ($W \cdot m^{-2} \cdot K^{-1}$),

Pr – Prandtl's criterion (1);

Indexes "w" apply to water, indexes "g" to apply to natural gas.

For the calculated speed of water circulation, it is necessary to finally check whether the water flow is also turbulent in the model. If, for example, at the compressor station, natural gas flows at a pressure of 7.35 MPa and a temperature of 60 °C through cooler tubes with an internal diameter of 25 mm at a speed of 8.05 $m \cdot s^{-1}$, then the coefficient α has a value of 1,632 $W \cdot m^{-2} \cdot K^{-1}$. On the model through which 60°C warm water will flow, the same value of α is reached at a speed of 0.271 $m \cdot s^{-1}$. The Reynolds criterion will be equal to 11,350, so the recirculation will be turbulent.

V. CONCLUSION

The designed measuring device described in the article will make it possible to perform measurements both for crossed current and for double crossed current in various combinations of pipe arrangement (placed in rows one above the other and in alternating arrangement). The inlet (outlet) temperatures of both media will be measured just before the entrance (outlet) to the cooling space, as well as the media flow rates. Thermocouples will be used to measure the water temperature. By monitoring these temperatures, information will be obtained about the thermal performance of individual pipes, depending on their arrangement in the heat exchanger. Air temperatures will be measured just before the entrance (exit) to the cooling space and in the space between the pipes. The overall heat transfer coefficient *k* will be calculated from the measured values, which is a basic indicator of the intensity of heat exchange. The highest attainable overall heat transfer coefficient k will be sought through different combinations of pipe arrangement in the exchanger. The measured values will be compared with the numerical solution using FVM in the ANSYS CFX simulation tool.

ACKNOWLEDGEMENTS

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