MODELING OF THERMAL BEHAVIOR OF RAW NATURAL GAS AIR COOLERS

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Abstract. When gas is being prepared for a long-range transportation, it passes through air cooling units (ACUs) after compressing; there, hot gas passing through finned tubes is cooled with air streams. ACU's mode of operation shall ensure a certain value of gas temperature at the ACU's outlet. At that, when cooling raw gas, temperature distribution along all the tubes shall be known to prevent local hydrate formation. The paper proposes a mathematical model allowing one to obtain a thermal field distribution inside the ACU and study influence of various factors onto it.

1. Introduction
One of operations necessary when preparing gas for long-range transportation is its cooling after every stage of compressing. Cooling is provided by air cooling units (ACUs), which are several sections of finned tubular heat exchangers, through which air is pumped or circulated with one or two fans. The target temperature of gas cooling depends on a number of factors, such as process modes of the gas dehydration plant, composition and moisture content of the natural gas, ambient temperature, etc. For raw natural gas, this parameter is of critical importance, because under subzero ambient temperatures, conditions arise for hydrate formation inside the ACU's tubes. As a result of local hydrate plug formation, pipe blockage occurs and rupture of the heat exchanging tubes follows [1]. That is why it is very important to be able to predict temperature values at individual sections of heat exchanging tubes under a given combination of process parameters, properties of gas being cooled and ambient conditions.

Currently, a number of various mathematical models of ACU thermal behavior are used, but all of them have certain drawbacks.

This paper proposes a refined mathematical model of thermal behavior, which may be employed with the ACU control system.

2. Overview of known solutions and formulation of research objective
Operation of an ACU proceeds as follows. A process fluid requiring cooling, in this case raw natural gas, is delivered into the heat exchanger tubes. The heat is transferred from the gas to the tubes, then from the tubes to the fins and further to the air, which withdraws the heat from the heat exchanger to the environment. The heat exchange may happen either by convective air flow of gravity-type through the heat exchanger, or by forced circulation by means of fans. At that, depending on the design, the fans either deliver the air to the heat exchanger, or draw it away.

Despite structural variation between different ACUs, their thermal behavior is closely linked to the
gas dynamic mode, because gas flow though each ACU depends on the average temperature inside the apparatus, which, in its turn, depends on the gas flowrate and air flowrate values. Thus, the thermal behavior modeling task is a non-linear problem.

Currently, ACU thermal behavior is calculated in accordance with one of the following three methods.

The method of ACU thermal behavior calculation proposed by VNIIGAZ is based upon analysis of a non-linear thermal balance equation by improving iterations for gas outlet temperature, which is first taken as an approximate value [2]. Within this model, a check-up correspondence analysis is performed to check the conformance between the calculated fan capacity and characteristics of the actual ACU fan. A value of static pressure is determined to assess the calculated power of the fan.

The main drawback of the VNIIGAZ method is impossibility to apply it to gravity-type convective flow, because the mean air temperature in the ACU section is calculated from a non-zero fan head [3]. Besides that, the method does not work when the values of the outlet gas temperature and ambient temperature are close to each other.

The TSPGU method differs in that when analyzing an ACU plant it is taken as a whole, not as separate ACUs [4]. At that, it is assumed that the natural gas flow is uniformly distributed between all the apparatuses, which is true only in case their geometry, physical characteristics and mode of operation are the same.

When a different number of fans operate on each ACU, distortions in the thermal behavior lead to changes in the gas dynamic mode and vice versa. Thus, this peculiarity of the method shall deem its disadvantage.

To rectify it, the method proposes taking into account ACU efficiency depending on the number of active fans by introducing empiric coefficients. However, when using this approach, the accuracy of the calculated outlet gas temperature of the ACU plant depends on how much the actual operating conditions of the ACU differ from the conditions for which the coefficients were determined experimentally.

Besides, this method uses constant values of ACU heat transfer coefficient and thermal properties of gases, while they actually depend on the temperature of the heat-transfer medium and other factors.

The third method, based on the Shukhov equation, is widely applied to determine thermal behavior of various process equipment in the oil and gas industry, including determination of an ACU target cooling temperature [5]. The Shukhov equation is, principally, a heat balance equation under the conditions of stable ambient air temperature. Within the proposed algorithm of ACU thermal behavior analysis, the Shukhov equation is solved by the Newton's method, where derivative with respect to temperature is substituted with a finite difference. The algorithm may be used to assess the ACU outlet gas temperature in both forced and gravity-type convection.

Among the disadvantages of this method, one may name a somewhat underestimated value of the output gas temperature due to an assumption that the cooling air flow has a constant temperature, that is, it has heat capacity, and the fact that the finned surface of the tubes is left unaccounted for.

ACU thermal field may be obtained by direct physical measurements as well; however, it is very hard to implement. A single ACU would require at least 45 temperature sensors arranged in accordance with a rather complex scheme [6, 7].

Thus, the problem of development of a mathematical model to optimize ACU operation in both stationary operation modes, depending on gas flow, seasonal and daily fluctuations of ambient temperature, and dynamic operation modes to cover random fluctuations, is a very timely one.

The problem of ACU thermal behavior determination is formulated in the following way: from known gas dynamic parameters and natural gas temperature and the ACU inlet and controlling actions, determine the temperature of the natural gas at the ACU outlet. Besides, in case of raw gas cooling, as it was mentioned earlier, there is the urgent problem of hydrate formation inside the heat exchanging tubes at low temperatures. That is why it is necessary to have the complete information about the ACU's thermal field during the operation of ACU. The mathematical model under development shall perform the following tasks:
– give the necessary information on the outlet gas temperature depending on other parameters and external conditions;
– maintain the gas temperature above the critical conditions corresponding to the onset of hydrate formation, or predict the locations of hydrates.

3. Main part

3.1 Assumptions of the mathematical model
The ACU thermal behavior is understood as a distribution of gas temperature along the heat exchange tubes.

Design of the majority of ACUs is based on the common principle: gas cooling by flowing a transverse flow of air onto bundles of finned tubes. This approach allows creating a common mathematical model of the gas cooling process from a number of assumptions:

– the gas flowrate is the same in all the tubes of the ACU;
– the air flowrate is the same throughout the section of the heat exchanger, physical properties of the mediums remain unchanged;
– the heat transfer rate along the whole length of the apparatus is proportional to the temperature difference of the mediums;
– displacement of gas in the direction of the gas flow is insignificant and not accounted for.

In case of a gravity-type convection flow, the heat exchange direction is from the lower rows of tubes to the upper rows, in case of a forced convection – the same as direction of the air flow from the fans. In both cases, temperature correlation between neighboring tubes in a horizontal row is negligible, thus, the common thermal field may be assumed as a set of one-dimensional thermal fields. Further, the heat exchange may happen either on conditions of the internal problem, when the core of the flow is in the internal area with respect to a boundary layer covering the channel walls, or on conditions of the external problem, when the core of the flow is in the external area with respect to the boundary layer. Both forms of heat exchange take place in the ACU: the internal problem is the heat exchange of gas flow inside the finned tubes, while the external problem is the heat exchange during the transverse flow of air across a bundle of tubes.

High speeds of the gas flow determine the turbulent nature of the flow, while high heat transfer rate of the tube material allows one to ignore its inertia and consider the heat exchange process between the air flow and the gas flow characterizing intensity of the heat exchange with a certain reduced coefficient of heat transfer from gas to air.

During the modeling, a thermal field of gas is determined along the length of the tube from known ACU parameters, properties of gas and thermal behavior parameters. Such parameters as ID and OD of the tube, finned tube length, smooth and finned surface, surface multiplier of the apparatus, fan capacity, and some others are necessary for the ACU. To characterize the thermal behavior one shall know the flowrate of hot and cold heat exchange medium, ACU inlet gas temperature, heat transfer coefficients for gas, air, tube wall; heat conductivity coefficient for gas and air; temperature of forced air; similarity criteria for gas and air.

3.2 Mathematical model of a gas thermal distribution
As per Shukhov equation, distribution of gas temperature in a single ACU tube with some assumptions may be determined in the following way:

\[
t_g = t_a + (t_{g in} - t_a) \exp \left( - \frac{\pi k (t_{g in} - t_a) S}{\rho_g C_g (t_{g in} - t_a)} \right),
\]

where \( t_a \) is the ambient air temperature, °C; \( t_{g in} \) is the ACU inlet gas temperature, °C; \( \rho_g \) is gas density, kg/m³; \( C_g \) is volume flowrate of gas, m³/s; \( C_g \) is specific heat capacitance of gas, (J·kg)/K; \( S \) is the area of the tube cross-section, m²; \( k \) is the heat transfer coefficient.

The main parameter characterizing the heat transferring ability of a surface is its heat transfer coef-
ficient, determined from the following formula:

\[ k = \frac{1}{\psi + \frac{1}{\alpha_g} + R_t \cdot \psi}, \tag{2} \]

where \( \alpha_g \) and \( \alpha_c \) are heat transfer coefficients from gas side and from cooling air side, respectively, \( W/(m^2 \cdot K) \); \( \psi \) is the surface multiplier of the apparatus; \( R_t \) is thermal resistance of possible contaminations inside the heat exchange tubes, \( (m^2 \cdot K)/W \).

Shukhov formula does not take into account two very important factors having direct influence on validity of the mathematical model: finned character of the heat exchange tubes and forced convective cooling (volume flowrates of air created by the first and the second fan), which will change the heat transfer coefficient.

A diagram of a finned heat exchange tube is shown in Figure 1.

\[ \psi = \frac{\varphi}{\alpha_g} \frac{\delta}{\lambda_p} + \frac{1}{\alpha_{pr}} + R_t \psi \]

where \( \varphi \) is the finned surface factor of the apparatus; \( \alpha_g \) is the heat transfer coefficient from gas, \( W/(m^2 \cdot K) \); \( \delta \) is the fin thickness, \( m \); \( \lambda_p \) is the heat conductivity of the fins, \( W/(m^2 \cdot K) \); \( \alpha_{pr} \) is the reduced heat transfer coefficient, \( W/(m^2 \cdot K) \).

The reduced heat transfer coefficient, accounting for heat transfer of the fin surface, smooth surface of the finned side and fin efficiency, is determined by the following formula:

\[ \alpha_{pr} = \frac{\alpha_g (F_{th} E + F_p)}{F_g}, \tag{4} \]

where \( E \) is the efficiency coefficient of the finned heat exchange surface; \( F_g \) is the total heat exchange surface of the finned side, \( m^2 \); \( \alpha_g \) is the heat transfer coefficient from air, \( W/(m^2 \cdot K) \); \( F_p \) is the finned surface area, \( m^2 \); \( F_g \) is the surface of the smooth area, \( m^2 \).

Distribution of gas temperature along a single finned tube of the ACU may be determined by the formula:

\[ t_g = t_v + (t_g \text{ in} - t_v) \cdot \exp \left( -\frac{k \cdot \pi d_{out}}{c_g p_g a_g} \cdot x \right), \tag{5} \]

where \( x \) is the coordinate along the tube, \( m \).

This formula suits for calculation of the natural gas thermal field under gravity-type convective flow in the ACU. To use this dependency to calculate forced convection temperature, the heat transfer coefficient shall be divided between two components \( k_1 \) and \( k_2 \), which depend on the volume flowrate.
of gas delivered by the first and the second fan respectively. As a result, a system of two equations is obtained, each of which calculates the gas temperature depending on the influence of fans, which change the head of the cooling air:

\[
\begin{align*}
    t_{g1}(x) &= t_v + \left( t_{g1}^{in} - t_v \right) \cdot \exp \left( -\frac{k_1 \cdot \pi \cdot d_{in}}{c_g \cdot p_g \cdot g_g} \cdot x \right), & \text{if } 0 \leq x \leq \frac{L}{2}, \\
    t_{g2}(x) &= t_v + \left( t_{g2}^{in} - t_v \right) \cdot \exp \left( -\frac{k_2 \cdot \pi \cdot d_{in}}{c_g \cdot p_g \cdot g_g} \cdot \left( x - \frac{L}{2} \right) \right), & \text{if } \frac{L}{2} < x \leq L
\end{align*}
\]

(6)

where \( k_1 = k(G_{g1}) \) and \( k_2 = k(G_{g2}) \) are heat transfer coefficients, depending on volume flowrate of the air delivered by the first and the second fan respectively; \( L \) is the tube length, m.

From the system of equations (6) by substituting the boundary condition in each of the equations \((x = \frac{L}{2} \text{ and } x = L)\), one may obtain an equation, determining the gas temperature at the outlet of a single ACU section:

\[
t_g(L) = t_v + \left[ t_v + \left( t_{g1}^{in} - t_v \right) \cdot \exp \left( -\frac{k_1 \cdot \pi \cdot d_{in}}{c_g \cdot p_g \cdot g_g} \cdot \frac{L}{2} \right) \right] \cdot \exp \left( -\frac{k_2 \cdot \pi \cdot d_{in}}{c_g \cdot p_g \cdot g_g} \cdot \frac{L}{2} \right).
\]

After certain simplifications, the thermal distribution of gas at the ACU outlet may be represented by the following expression:

\[
t_{g, out} = t_v + \left( t_{g}^{in} - t_v \right) \cdot \exp \left[ -\frac{\pi \cdot d_{in}}{c_g \cdot p_g \cdot g_g} \cdot \frac{L}{2} \cdot (k_1 + k_2) \right].
\]

(7)

This mathematical model describes the gas thermal field at the outlet of the ACU with two consecutive fans.

### 3.3 Verification of the mathematical model

Verification of the mathematical model of the tube thermal field was conducted for a concrete type of ACU, namely, 2AVG-75, in the Mathcad Prime software package. The following values of input and control variables were used: ambient air temperature from 0 to minus 40 °C; ACU inlet gas temperature is 70 °C; gas density is 0.68 kg/m³; volume flowrate of gas is 0.23 m³/s; volume flow rate of air at the first and the second fan is from 0 to 100 m³/s.

The results of modeling the thermal field and thermal behavior of the ACU outlet gas are shown in Figures 2 and 3 respectively. In Figure 3, surface regions denote the ACU outlet gas thermal field for possible ranges of change in volume flowrate of the first and the second fan. Each region differs by the ambient air temperature, starting (from the top) from 0 down to minus 40 °C. The yellow region denotes the onset of gas condensation, which proceeds until the red region. The red region denotes the onset of hydrate formation in the ACU heat exchange tubes.

Validity of the proposed mathematical model was assessed by comparing the calculated temperature values with the actual values at the outlet of a 2AVG-75 ACU installed in the Urengoy oil and gas condensate field (at maximum rpm of both fans) for the ambient temperatures from minus 25 to 30 °C. Maximum absolute error was 1.5°C; maximum relative error was 6.15%, which may be deemed acceptable.
4. Conclusions
Valid information on current gas temperature at the ACU outlet and thermal field at all the sections of the finned tubes is necessary for efficient operation of the ACU and prevention of local hydrate formation inside. Provision of ACU with hardware sensors is labor-intensive and impractical, thus a mathematical model is necessary, which may adequately reflect temperature distribution inside the ACU in both static mode of operation and in dynamic conditions arising due to various perturbations. The proposed model is based on the Shukhov equation and allows determining the temperature at any section of every ACU for all the modes of fan operation, including gravity-type flow. Validity of the model underwent successful experimental testing.

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