Model for Simulating the Temperature Conditions of a Sour Gas Cooling Unit

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Abstract. While being prepared for long-distance transport, gas is subjected to multiple cooling procedures that use air-cooling units. This is necessary for reducing the gas volume and preventing the gas from affecting the pipeline and the permafrozen soil where the pipeline is laid. ACU control systems are mainly tasked to sustain the necessary gas temperature at the unit outlet, which must provide sufficient cooling while also preventing the generation of hydrates in the ACU tubes. The ACU outlet temperature is the mean temperature, as temperatures of individual outlets might vary greatly. Therefore, normal ACU operation requires a mathematical thermal field model that could be used to find the temperature at any point of the pipe. The paper proposes a thermal field simulation model that can be used to find temperatures at specific points of heat-transfer pipes as a function of the ambient temperature for any fan-vs-fan performance ratio. Experimental verification of the model showed good repeatability.

1. Introduction
State-of-the-art gas production is a complex branched structure that includes various facilities such as gas-well clumps, the gas transport network, the gas booster stations, gas conditioning units, etc. The eigen-energy of gas may not be sufficient for transporting it from the well via the GTN to the GCU, which is why it is additionally compressed at the GBS. This increases both the temperature and the pressure. Temperature at the outlets depends on the initial temperature at the GBS inlets as well as on the gas compression. Higher temperatures result in various undesirable effects such as changes in the thermobaric conditions of gas dehydration at the GCU, deterioration of the gas line insulation, thawing of the permafrost (most of the gas fields are located in the Far North), and greater energy consumption for gas compression due to increased volumetric flow.

In order to cool the compressed gas down, air-cooling units are installed at the outlet of each compression stage. The temperature to which the gas must be cooled down depends on the GCU parameters.

The ACU are tubular heat exchangers that cool the gas down by means of heat exchange with the air flow [1]. Depending on the heat-transfer method, an ACU can utilize natural air convection via the heat exchanger, or fan-forced air circulation. An ACU consists of finned heat-transfer pipes, driven fans, diffusers and louvers, supporting structures, and control mechanisms. One obvious advantage of ACUs is that they keep the cooled medium pure thanks to closed loops; besides, they can be installed in virtually any climate or natural area, are not too expensive to operate, and are eco-friendly. However, certain restrictions apply to the use of ACUs, one of them being their unsuitability for non-dehydrated natural gas at negative ambient temperatures since such conditions may result in the...
generation of hydrates in the heat-transfer tubes. Hydrate generation results in the blockade of heat-transfer pipes, entailing their imminent destruction. This is why controlling the gas temperature in the ACU is the main problem that ACU control systems are intended to solve. Temperature is mostly controlled by altering the amount of cooling air fed for heat transfer [2]. Air feed can be controlled by louvering the heat-transfer surfaces and the fan outlet, altering the blade rotation angle using variable-speed motors, or switching off fan or heat-transfer surfaces. Anyway, control is based on the gas temperature at the ACU outlet, which is the mean gas flow temperature of all tubes. It is therefore possible that while the mean temperature may be within the acceptable parameters (i.e. will not result in hydrate generation), actual tube-specific temperatures might deviate [3]. Paper [4] notes that hydrate generation is the most intense in the bottom row of heat-transfer pipes (in the direction of cooling-air flow). This is why developing a model to simulate the thermal field of an ACU is a relevant problem of applied science.

2. State-of-the-Art and Statement of Problem

Various methods can be used to simulate and calculate the ACU thermal field. For example, Shukhov’s formula is used to compute the natural-gas temperature field for unrestricted-convection ACUs [5]. This formula is essentially a thermal-equilibrium equation for invariable ambient air temperature.

Therefore, the gas temperature distribution equation for a single ACU pipe can be written as following under certain assumptions:

\[ t_g = t_v + t_g^{\text{in}} - t_v \exp \left(-\frac{\pi \cdot k \cdot t_g^{\text{in}} - t_v}{p_g \cdot G_g \cdot c_g \cdot t_g^{\text{in}} - t_v} \right) \]

where \( t_v \) is the ambient air temperature, °C; \( t_g^{\text{in}} \) is the ACU-inlet gas temperature, °C; \( p_g \) is the gas density, kg/m³; \( G_g \) is the volumetric gas flow, m³/s; \( c_g \) is the specific heat capacity, (J/°C/kg); \( S \) is the tube cross-section, m²; \( k \) is the heat-transfer coefficient.

Heat transfer coefficient \( k \) is found as follows:

\[ k = \frac{\psi \cdot 1}{\alpha_g + \frac{1}{\alpha_e} + R_s \cdot \psi} \]

Where \( \alpha_g \) and \( \alpha_e \) are heat transfer coefficients for gas and for the cooling air, respectively; W/(m²·K); \( \psi \) is the unit surface enlargement factor; \( R_s \) is the thermal resistance of contaminants that might be contained in the heat-transfer pipes (m²·K)/W.

In [6], Shukhov’s equation is used to compute the gas temperature distribution along the pipe (Fig. 1) for an ambient-air temperature of 8 °C negative, and a gas flow of 1,200 m³/h (via the ACU).

The graph is non-linear, as the difference between cooling-air and cooled-gas temperatures peaks at the beginning of a heat-transfer section, resulting in the most intense heat transfer; further along the section, the gas becomes cooler, entailing less intensive heat transfer.

Shukhov’s formula can be used to evaluate how the inner diameter of the heat-exchanger pipe and the ACU gas flow affect the gas temperature, all other things constant. Figure 2 presents a Shukhov-based graph of the gas temperature and pipe ID correlation (the ID changes from 0.015 to 0.028 meters with a 0.001-m pitch); Figure 3 presents a graph of the gas temperature and ACU gas flow correlation (flow values from 0.1 to 0.3 m³/s with a 0.02-m³/s pitch).
Figure 1. Gas temperature distribution in the ACU along the heat-transfer section: plotted by Shukhov’s equation

Figure 2. Gas temperature and pipe ID correlation. Legend (left to right): gas temperature (degrees Celsius), pipe length (m), pipe diameter (m)
Figure 3. Temperature dependence of gas along the length of the pipe from the gas flow. **Legend (left to right):** gas temperature (°C), pipe length (m), gas flow rate (m³/sec)

For convection, the Shukhov-based model is consistent with real-world experimental results. For instance, for the temperature-pipe ID correlation graph, the variance is within 1.5%; for temperature-flow correlation, it is within 1.3%.

However, in case of forced heat transfer, Shukhov’s formula neglects two very important factors that directly affect the adequacy of the mathematical model: finning of heat-transfer pipe and forced convective cooling (volumetric air flow induced by Fan 1 and Fan 2); these factors alter the heat-transfer coefficient.

There is a large number of research papers whose subject is the ACU thermal field for various operating conditions [6-12]. For instance, in [6] a single-fan ACU is modeled based on an assumption that air flow from a single fan is evenly distributed across the entire section area. This approach might result in unacceptable error. In fact, ACU design implies that fans are fully separated in terms of ducting. This is done to prevent the loss of air blown by a single operating fan, which could otherwise go to the duct of the switched-off fan and bypass the heat-transfer section; that said, each fan can blow only one half of the section. For such cases, paper [5] proposes modeling an ACU with a single operating fan by making calculations for a half-length section blown by a single operating fan, then computing a half-length section in an unrestricted-convection mode.

The objective hereof is to develop a model to simulate the thermal field of a dual-fan ACU with any arbitrary air flow per fan.

3. **Results and Discussion**

Shukhov’s equation can be used for calculating the temperature in forced convection; however, the heat coefficient must be divided into two components $k_1$ and $k_2$, which are functions of the volumetric gas flow as supplied by Fan 1 and Fan 2, respectively. Each fan is VFD-controllable by altering the rotation speed which changes the cooling-air pressure. As a result, we obtain a dual-equation system, where each equation calculates the gas temperature as a function of fan effects:
\[ t_{g1}(x) = t_v + t_v^{\text{in}} - t_v \exp\left(-\frac{\pi \cdot k_1 \cdot d_{\text{in}} \cdot x}{p_g \cdot c_g \cdot G_g}\right) \quad \text{at} \quad 0 \leq x \leq \frac{L}{2} \]

\[ t_{g2}(x) = t_v + \left(t_{g1} \left(\frac{L}{2}\right) - t_v\right) \exp\left(-\frac{\pi \cdot k_1 \cdot d_{\text{in}} \cdot \left(x - \frac{L}{2}\right)}{p_g \cdot c_g \cdot G_g}\right) \quad \text{at} \quad \frac{L}{2} < x \leq L \quad (3) \]

where \( k_1 = k \cdot G_{v1} \) is the heat transfer factor depending on the volumetric air flow, Fan 1;
\( k_2 = k \cdot G_{v2} \) is the heat transfer factor depending on the volumetric air flow, Fan 2;
\( L \) is the pipe length, m.

To produce an equation to find the gas temperature at the outlet of a single ACU section, substitute \( x = \frac{L}{2} \) in the first equation of the system, and \( x = L \) in the second equation.

Combine the equation system (3) with Shukhov’s formula (1) to get:

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

After some simplifications, temperature distribution of gas at the ACU outlet with forced airflow will be described as follows:

\[ t_{g}^{\text{in}} = t_v + t_v + t_v^{\text{in}} - t_v \exp\left(-\frac{\pi \cdot d_{\text{in}} \cdot \frac{L}{2}}{p_g \cdot c_g \cdot G_g} \cdot k_1 + k_2\right). \quad (5) \]

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

The model has been applied to 2AVG-75-type ACUs that are widely used in gas condensate fields, including the Urengoy gas field.

Simulation model was generated in Mathcad Prime.

Figure 4 shows the presentation of analytical dependencies for the temperature field of the ACU pipe, as well as the equation system presentation.
The model contains a set of internal parameters plus input/control/output variables. The internal parameters of the model are the specifications of the 2АВГ-75 unit, such as the area of heat-transfer surface of finned pipes, the finning factor, the surface enlargement factor, the number of pipes per section, the number of pipes, the internal diameter of the supporting pipe, the outer diameter of the supporting pipe, the rated rotation speed and performance of fans etc. The input variables are the ambient air temperature and gas temperature at the ACU outlet, the volumetric gas flow, the gas density, pressure, and humidity; the control variables are the volume air flows for Fan 1 and Fan 2; the output variable is the ACU outlet gas temperature.

The model has been tested for its ability to show correlations of gas temperature and dual-fan air flows. Figure 5 shows the values of input and control variables.

### Input variables
- Ambient temperature: \( t_v = -30 \)
- Gas temperature on an entrance to ABO: \( t_{vx} = 70 \)
- Gas density: \( \rho = 0.68 \)
- Volume flow of gas: \( G_v = -0.23 \)

### The operating variables:
- Volume flow of air on the first fan: \( G_{v1} = -0\ldots100 \)
- Volume flow of air on the second fan: \( G_{v2} = -0\ldots100 \)

**Figure 5.** Input and control variables in Mathcad Prime. **Legend (top to bottom): input variables,** ambient air temperature, gas temperature when entering ACU, gas density, volumetric flow rate, **control variables,** volumetric flow rate at Fan 1, volumetric flow rate at Fan 2

The obtained calculated correlations are shown in Figure 6.

\[
t_1(x, G_{v1}) = t_v + t_{vx} - t_v \cdot \exp \left( \frac{-0.017 \cdot x}{G_{v1}^{0.68} + 0.109} \right)
\]

\[
t_2(x, G_{v1}) = t_v + \left( t_{vx} - t_v \cdot e^{-0.102 G_{v1}^{0.68} + 0.109} \right) \cdot e^{-0.017 \cdot x - 6 \cdot G_{v2}^{0.68} + 0.109}
\]

\[
t_2(x, G_{v2}) = \begin{cases} 
\frac{L}{2} & \text{if } 0 \leq x \leq \frac{L}{2} \\
\frac{L}{2} & \text{if } \frac{L}{2} \leq x \leq L \\
error & \text{else}
\end{cases}
\]

**Figure 6.** Presentation of the calculated correlation in Mathcad Prime

Calculation results are shown in Figures 7 and 8.
Any model is deemed adequate to the object if theoretical computations are consistent with the experiment within the latter’s margin of error [13].
To evaluate the error of our simulation model, we have calculated the absolute and relative error. The actual gas temperature at the ACU outlet was measured at an actual unit at maximum rotation speed of both fans, with all parameters except the ambient air temperature remaining constant.

Figure 2 shows the deviations of ACU outlet temperatures, modeled vs real data.

![Figure 9. Deviations in modeled and real temperature data as correlated to the ambient temperature](image)

Maximum relative error amounted to 6.15%.

4. Conclusions
The resulting model for simulating the temperature conditions of sour-gas ACUs can be used to analyze the unit’s operation when exposed to various factors, and to obtain the necessary correlations of gas temperatures and temperature-affecting parametric changes. The model-derived temperature distribution of the gas along the heat transfer pipe enables one to pre-determine the areas of possible condensate deposition and hydrate generation, as well as the conditions under which hydrates may be generated. This allows one to choose such fan parameters as to avoid that.

This is why the simulation model can be further used to compute the gas ACU control and regulation algorithms for a more efficient cooling and prevention of hydrate generation.

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