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FLOW AND HEAT TRANSFER INSIDE A NEW DIVERSION-TYPE GAS HEATING DEVICE


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ABSTRACT

The present paper characterizes ethylene glycol flow and heat transfer inside a new diversion-type gas heating device. A 2-D natural convection heat transfer model was built and solved by the finite volume method with unstructured body-fitted grids. The numerical model was first validated through temperature comparison with experimental measurements in a conventional device structure. Then analyses and comparisons of the flow fields and temperature distributions with use of different guide plate structures were carried out. The numerical results show that using the guide plate structures can form better organized flow patterns that augment heat transfer. The heat required for heating up the gas passing through the heating device can be reduced by 3% via installing two guide plates.
NOMENCLATURE

\( b \)  
Spacing, m

\( c_p \)  
Specific heat capacity, J kg\(^{-1}\) K\(^{-1}\)

\( d \)  
Diameter of convective tube, m

\( D \)  
Diameter, m

\( h \)  
Heat transfer coefficient, W m\(^{-2}\) K\(^{-1}\)

\( k \)  
Thermal conductivity, W m\(^{-2}\) K\(^{-1}\)

\( L \)  
Length, m

\( n \)  
Normal direction

\( Nu \)  
Nusselt number

\( p \)  
Pressure, Pa

\( Pr \)  
Prandtl number

\( Q \)  
Total heat transfer rate, W m\(^{-1}\)

\( \text{Re} \)  
Reynolds number

\( t \)  
Thickness, m

\( T \)  
Temperature, K

\( u, v \)  
Velocity components, m s\(^{-1}\)

\( x, y \)  
Coordinates, m

Greek Symbols

\( \beta \)  
Thermal expansion coefficient, K\(^{-1}\)

\( \mu \)  
Dynamic viscosity, kg m\(^{-1}\) s\(^{-1}\)

\( \phi \)  
Angle between the convective tubes, °

\( \rho \)  
Density, kg m\(^{-3}\)

Subscripts

\( \infty \)  
Reference value

\( e \)  
Ethylene glycol
1. INTRODUCTION

Natural gas, as a clean energy, has been widely used in power generation, combustion and fuels, residential and commercial buildings, to name a few. The rapid development of natural gas industry is bound to cause and promote the development of the related equipment and technologies [1, 2]. During transportation of natural gas, the containing hydrate will separate out and coagulate as temperature falls, causing potential blockage of pit shafts, pipes, valves and equipment [3, 4]. To prevent such a disaster, heating of gas during transportation is needed [5]. On the other side, gas temperature decreases as pressure drops during use of natural gas. To meet the combustion needs and improve the combustion efficiency, preheating of gas is usually needed as well [6]. Further, heating is also needed to gasify and heat up liquefied natural gas (LNG) before being used in LNG transmission and application systems, especially in the relatively independent areas where a natural gas supplying network is not available, and in the regions with large load variation range such as in industries that require frequently quick gas supply or stop [7]. Therefore, heating process is an essential link in the whole natural gas application processes. A heating device is an indispensable piece of equipment in gas industry. It plays the role of heating up natural gas to the required processing temperature and the application of gas heating device shows a promising prospect [8].

Since natural gas and LNG are flammable and explosive, National Standards [9] in China demand that it be heated indirectly by flame or flue gas [10]. The types of gas heating device that have been considered mainly include heaters with heating finned tubes [11], submerged combustion heaters [12, 13], cylinder-type heaters with an intermediate heating medium [14], and heaters with seawater as the heating medium [15,}

\[ g \quad \text{Gas} \]
\[ w \quad \text{Tube wall} \]
Among them, the cylinder-type heating device with an intermediate heating medium is most widely employed [17, 18].

The cylinder-type heating device adopts a solid assembled structure, as shown in Fig. 1. Generally, the heating surfaces (incl. the fire tube and flue tube bundle) and the cooling surfaces (i.e., the convective tube bundle) are symmetrically arranged via the central axis of the circular cross-section placed inside a large cylinder. The heating surfaces are placed below the horizontal axis and the cooling surfaces are placed above the horizontal axis. The cylinder is filled with a heat-transfer medium, such as water [19, 20] or ethylene glycol [21, 22]. Heat from the fuel combustion is transferred to the heat-transfer medium through the fire tube wall and the flue tube bundle wall. At the same time, the heat-transfer medium transfers most of the heat to the gas or LNG inside the convective tube bundle. The heat-transfer medium in the cylinder flows due to buoyancy generated by the uneven density distribution caused by the temperature difference.

Natural convective heat transfer occurs between the heat-transfer medium and the heating/cooling surfaces. However, some defects exist in the conventional arrangement of heat-exchange surfaces in such gas heating devices. For example, the temperature of the heating medium close to the wall of the fire tube and flue tube bundle is higher and the medium fluid in this lower region of the cylinder forms an uprising flow; while the temperature of the medium close to the convective tube bundle is lower and the medium in this upper region forms a descending flow. Because there is no channel to separate the flows, these two counter-opposed flows impact each other. Additionally, there exist temperature differences between the fire tube and the flue tube bundle and between the multi-return passes of the convective tube bundle. All these can cause chaos in the heat-transfer medium fluid flow and result in a poor flow pattern and uneven heating [23].

Moreover, the driving force for the natural convection is small due to the small temperature difference between the up and low parts of the flow field. The flow formed could hardly be effective such that the convective heat transfer is very weak between the
heating and cooling surfaces. In some cases, the temperature difference between the cold and hot fluids is negligible, such that the flow field in the cylinder may even be considered in a static state and the heat transfer mode would be close to heat conduction [24]. It results in a very low heating efficiency in the heating device.

The working temperature range of ethylene glycol is larger than that of water due to its high boiling point and low vapor pressure [25]. On the other side, use of ethylene glycol as the heating medium weakens the action of buoyancy force because of its big viscosity. This might form some flow dead angles. If the design of the heat transfer component inside the gas heating device can be improved, however, use of ethylene glycol could widen application ranges in addition to improve thermal efficiency [26]. Addition of metallic plates [27] was a common approach for enhancing heat transfer.

Here we propose to augment heat transfer inside a gas heating cylinder with use of metallic guide plates. Viscous ethylene glycol is employed as the heating medium to widen the temperature range of applications. Depending on the different usage conditions and the technical requirements, the guide plates can be designed with different shapes and arrangements to fit into different applications [28]. The shape of the guide plates can be designed as flat or arc and the guide plate can be arranged in one side or two sides. In this study, we focus on numerical investigation of the effects of different guide plate arrangements on fluid flow and heat transfer. The temperatures simulated are compared with experimental measurements at some positions. Three types of guide plate configuration are considered and compared.

2. PHYSICAL MODEL AND EXPERIMENTAL SETUP

The experimental system is shown in Fig. 2. Its main body is a gas heating device with dimensions and specifications listed in Table 1. An insulating layer about 8mm thick is applied to wrap the outside of the large cylinder. In order to test the temperature distribution of the heat-transfer medium inside the cylinder, nine thermocouples are placed
in the cross-section A-A as shown in Fig. 2(b). Measuring point 0 is located in the center of the circular section and is defined as the central temperature of the cylinder. Measuring point 1 is located at the third pass region of the convective tube bundle which is in the left side of the cylinder. Measuring point 2 is at the fourth pass region of the convective tube bundle which is in the left side of the cylinder. Measuring point 3 is at the second tube pass region in the cylinder’s right side. Measuring point 4 is at the first tube pass region in the cylinder’s right side. Measuring point 5 is arranged in the upper part of the flue tube and measuring point 6 is in the lower part of the flue tube. Measuring point 7 is in the upper part of the fire tube. Measuring point 8 is at the bottom of the cylinder. The uncertainty of temperature measurement is determined mainly by two factors – the accuracy of the thermocouples and uncertainty of placement of the thermocouples. To compare with the numerical results later, we estimate the uncertainty of the measuring temperature is ±2 K. Many tests were conducted. The repeatability and consistency of the measured central temperature were good. This temperature can be used as a measure justifying the quasi steady state during the operation of the heating device. During experiment, the fuel rate was set at 8 m³/h and the flow rate of the heated gas was set at 1.67 m/s; the pressure of the heated gas was 5.5 MPa, and the inlet and outlet temperatures of the heated gas were 280 K and 320 K, respectively.

3. MATHEMATICAL MODEL

In the present numerical simulation, the end effects of the cylinder are neglected because the cylinder is relatively long and the flow and temperature in the middle section were found to be stable. A two-dimensional (2-D) circular cross-section of the experimental cylinder is adopted for simulation and the configuration detail is shown in Fig. 3. The dimensions and configurations of the numerical model are consistent with the experimental structure, except for that thin-wall assumption for the tubes inside is employed in simulation.
The horizontal and vertical distances between the heating and cooling surfaces, \( b_1 \) and \( b_2 \) shown in Fig. 3(a) are 540 mm and 510 mm, respectively. The diameter of the large cylinder \( D_1 \) is 1300 mm; the diameter of the fire tube \( D_2 \) is 325 mm. The flue tube bundle consists of 24 tubes of 42 mm in diameter, the spacing between the tubes is 68 mm. The convective tube bundle consists of 9 tubes of 38mm in diameter and it has four round trips. The spacing between the tubes is as shown in Fig. 3(b), where \( b_3 \) is 90 mm, \( b_4 \) is 65 mm, and the angle \( \varphi \) is 60°.

The physical properties during calculations were assumed to be constant, except for the density. The density is assumed to be linearly proportional to temperature because the medium temperature variation is small. The fluid flow is considered as a steady-state laminar flow, so that the viscous dissipation is negligible. The cylinder wall is treated as an adiabatic boundary and the heat losses there are ignored.

Based on the Boussinesq assumption under the steady-state laminar flow condition, the 2-D flow and heat transfer governing equations are listed below [29, 30].

Continuity equation:

\[
\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} = 0 \tag{1}
\]

Momentum equations:

\[
\rho(u \frac{\partial u}{\partial x} + v \frac{\partial u}{\partial y}) = -\frac{\partial p}{\partial x} + \mu \left( \frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\partial y^2} \right) \tag{2}
\]

\[
\rho(u \frac{\partial v}{\partial x} + v \frac{\partial v}{\partial y}) = -\frac{\partial p}{\partial y} + \mu \left( \frac{\partial^2 v}{\partial x^2} + \frac{\partial^2 v}{\partial y^2} \right) + \rho g \beta (T - T_\infty) \tag{3}
\]

Energy equation:

\[
\rho(u \frac{\partial T}{\partial x} + v \frac{\partial T}{\partial y}) = \frac{k}{c_p} \left( \frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} \right) \tag{4}
\]

In which, \( u, v \) are the velocity components on \( x, y \) directions, respectively; \( p \) is the pressure; \( \mu \) is the fluid dynamic viscosity; \( \beta \) is the fluid volume expansion coefficient; \( \rho \) is
the fluid density; $T$ is the temperature; $T_\infty$ is the fluid reference temperature; $k$ is the thermal conductivity; and $c_p$ is the specific heat.

In accordance with the actual operating conditions of a heating device, the calculation boundary conditions are specified as follows: the wall of the cylinder is treated as an adiabatic boundary; the average temperature of the fire tube wall is set as 420K, and the average temperature of the flue tube bundle wall is set as 380K; the boundary condition of the third kind is applied to the boundary of the convective tube bundle, and it is expressed as:

$$-k_e \frac{\partial T_w}{\partial n} = h(T_w - T_g)$$  \(5\)

where $k_e$ is the thermal conductivity of heat-transfer medium; $n$ indicates the wall normal direction; $T_w$ is the wall temperature; and $T_g$ is the heated gas temperature. The average temperature of the heated gas in the first and second tube passes is 290 K, and the average temperature of the heated gas in the third and fourth tube passes is 310 K. $h$ is the heat transfer coefficient calculated by

$$h = \frac{N\varepsilon \cdot k_g}{d}$$  \(6\)

in which $k_g$ is the thermal conductivity of gas inside the convective tube bundle; $d$ is the diameter of the convective tubes; and the Nusselt number is calculated by a correlation [31]:

$$Nu = 0.023 \text{Re}^{0.8} \text{Pr}^{0.4}$$  \(7\)

where $\text{Re}$ is the Reynolds number for the gas flow inside the convective tubes, and $\text{Pr}$ is the gas flow Prandtl number.

Based on the finite volume method, the commercial software package Fluent 6.3 is used in the present simulations of the fluid flow and heat transfer in the cylinder. The second-order windward difference scheme is adopted for the convection diffusion terms and the pressure and velocity are coupled by a simple algorithm.
The body-fitted solution with unstructured Cartesian grids is adopted [32]. Not only the generated grid system for the irregular heat-exchanger surfaces is simple and intuitive [33], but also the conversions of node coordinates and the governing equations in the physical and computational domain are avoided. Mesh refinement is necessary in the boundary surface vicinity, such as near the heating and cooling surfaces where strong velocity and temperature gradients exist. A typical grid system used in computation is plotted in Fig. 4.

Grid-independency is examined with five different grid levels: 132, 255, 399, 837, and 1,260 thousands cells. Fig. 5 compares the computed total heat transfer rate in the convective tube bundle (i.e., the heat used to heat up gas inside the convective tubes) and the temperature at the center of the cylinder in the case without guide plate. It is seen that the total heat transfer rate as well as the central temperature converges with refining of the grid. The difference between the finest grid having 1,260 thousands cells and the grid having 399 thousands cells is less than 1.2% for the total heat transfer rate, and less than 0.13% for the temperature at the cylinder center. Therefore, the setting of the grid level of the latter (399 thousands cells) was used for all the simulations thereafter.

4. RESULTS AND DISCUSSION

Before proceeding to investigate the influence of the guide plates, the temperatures simulated are compared with the experiemental measurements at the 9 locations marked in Fig. 2(b) for the conventional cylinder without use of guide plate. The comparison is shown in Fig. 6. It is seen that the simulated temperatures are in good agreement with the experimental measurements. The simulated temperatures are generally 1~2.5°C greater than the corresponding temperatures experimentally measured due to the neglection of axial flow and heat dissipation through the cylinder shell in the simulation.

Refering to the locations of temperature measurement, a clear thermal stratification at different heights of cylinder’s left and right regions is observed according to the
temperature distribution pattern. The temperatures are higher at measuring points 5 to 7 which are closer to the fire tube and flue tube bundle. The highest temperature appears at measuring point 7 which is the nearest to the fire tube. Small temperature differences exist among the measuring points 1 to 4 located at different convective tube pass regions. The temperature at measuring point 8 located at the bottom of the cylinder is the lowest.

Moreover, the temperature difference between the upper and lower parts of the flow field is insignificant. Thus, the driving force causing natural convection is weak in this traditional cylinder arrangement. It took about 4 hours for the test to reach a thermal equilibrium condition in the experiment. The start-up of the heating device was very slow. This indicates the necessity to enhance the heat transfer inside the cylinder with modified structures, such as usage of guide plates. In the followings, the fluid flow and heat transfer with use of guide plates are simulated. The guide plates are made of carbon steel. The heating medium is still ethylene glycol.

The temperature and flow fields for the case with a single flat guide plate are shown in Figs. 7 (a) and (b), respectively. The guide plate is 350 mm long and 6 mm thick. One side of the guide plate is tangent to the right side of the fire tube and the other side is connected with the convective tube at the left bottom side of the first pass of the convective tube bundle. As seen from the temperature distribution in Fig. 7(a), a long narrow high temperature zone clings to the left side of the guide plate, while a high temperature horn-shaped zone appears on both sides of the flue tube bundle, which points to both sides of the channel of the above cooling surface. An obvious descending flow trend is formed around the first and second passes of the convective tube bundle and the right side of the guide plate appears a lower temperature down-flow zone.

From the flow vectogram in Fig. 7(b), it is seen that the guide plate blocks the channel between the fire tube and the first pass of the convective tube bundle. As a result, the ethylene glycol releases heat when it flows through the first and second passes of the convective tube bundle along the guide plate after absorbing heat from the fire tube and
flue tube bundle below, and then it flows back to the fire tube along the other side of the guide plate. A big flow circulation is formed. Because of no guide plate at the left side, the ethylene glycol around the flue tube bundle lacks effective lead, which causes a local hedging with the descending flow around the above convective tube bundle.

A guide plate can be installed for not only totally blocking, but also partly blocking. In order to obtain a smoother flow field, an arc guide plate might be a better choice. Figs. 8 (a) and (b) show the temperature and flow fields, respectively, for the cylinder with a single arc guide plate. The arc guide plate of 320 mm long and 6 mm thick is placed between the fire tube and the convective tube bundle. The plate is tangent to the right side of the fire tube and extends to the top left. The channel between the fire tube and the first pass of the convective tube bundle is partly blocked.

It is seen from Fig. 8(a) that a narrow arc high temperature band appears on the top of the fire tube and a high temperature horn-shaped zone also appears on both sides of the flue tube bundle, which points to both sides of the channel of the cooling surface above. An obvious descending flow trend is formed in the surrounding of the first and second passes of the convective tube bundle and the right side of the guide plate presents a lower temperature down-stream zone. In Fig. 8(b), it is seen that the ethylene glycol around the fire tube flows upward along the curved guide plate, and then mixes with the ethylene glycol around the flue tube bundle. It flows through the multi-return of the convective tube bundle in turn. After the heat release process, the ethylene glycol will drop and reflow back to near the fire tube along the other side of the arc guide plate. A clockwise and smoother large flow circulation is formed, due to the better pertinence between the arc guide plate and streamline of the heat flow. Therefore, the arc plate enhances heat transfer.

In some situations, the thermal efficiency with a single plate is still poor if the thermal pressure between the upper and lower parts of the flow field is small. It could install two guide plates. Figs. 9 (a) and (b) show the temperature contours and velocity vector maps, respectively, for the case with double guide plates. One flat guide plate is set at the place
between the right side of the fire tube and the left side of the first pass of the convective tube bundle. Another one is placed between the left side of the flue tube bundle and the right side of the fourth pass of the convective tube bundle. Both plates are about 350 mm long and 6 mm thick. As seen from the temperature contour in Fig. 9(a), the narrow high temperature zone on the top of the fire tube is close to the right side guide plate, while the high temperature zone above the flue tube bundle is close to the left side guide plate. A down welling trend is formed at the bottom right of the first and second passes of the convective tube bundle and a lower temperature down-stream region appeared. Moreover, a down welling trend is also formed at the bottom left of the third and fourth passes of the convective tube bundle, and a lower temperature down-stream zone also appeared. As seen from the velocity vector map in Fig. 9(b), the ethylene glycol around the fire tube goes upward and flows through the convective tube bundle in the right region, and then comes back to near the fire tube. The ethylene glycol around the flue tube goes upward and flows through the convective tube bundle in the left region, and then comes back to near the flue tube bundle. Hence, a double-circulation flow field of circumfluence from center to two-side is formed on the circular cross-section in the cylinder.

No matter what kind of arrangement is adopted, the guide plates separate the flow region between the heating and cooling surfaces reasonably. Obviously, the guide plates make the thermal boundary layer close to the wall thinner, guide the heat-transfer medium to form an overall organized smooth flow; and thus, the flow dead angles easily formed in conventional cylinder without guide plate are eliminated.

The effects of guide plate on the total heat transfer rates in the convective tube bundle and the temperature at the measuring points around the convective tube bundle are displayed in Fig. 10. It is seen that the temperatures of ethylene glycol around the convective tube bundle without use of guide plate are significantly higher than those with use of a single or double guide plate(s). There also exist certain differences in the temperature distribution around the multi-return passes of the convective tube bundle.
between the single guide plate and two guide plates, due to the different shapes and arrangements of guide plates. When heating up the heated gas to the required temperature, the total heat transfer rate of the convective tube bundle can be reduced 1.7~3% via installing guide plates. The total heat transfer rate in the convective tube bundle for the two side guide plates is the lowest. Such a diversion-type heat transfer mode is especially suitable for gas heating devices with heat-transfer medium of high viscosity such as ethylene glycol considered in this study.

5. CONCLUSIONS

(1) To overcome the weakness that an effective heating flow field cannot be formed inside the conventional gas heating device, use of various guide plates installed between the heating and cooling surfaces to improve the overall heat transfer efficiency is considered. The simulated results in this study demonstrated the feasibility and advantages of such a heat transfer structure within a gas heating device.

(2) Installing guide plates along the axial direction of the large cylinder is a simple and effective heat transfer enhancement method, especially for high viscosity heating medium such as ethylene glycol. An effective and smooth flow field can be formed in the modified gas heating device.

(3) The guide plates can be designed with different shapes and arrangements to adapt different engineering applications. The total heat transfer rate in the convective tube bundle can be reduced by 1.7~3%, with the use of double guide plates is the most efficient among the three considered cases.

(4) The diversion-type gas heating device has the merits of simple structure, wide application range, and quick start, etc. The cost for installation of guide plates is low and can be recouped in a short time. The new structure can be used not only in the design of a new heating device, but also in the transformation of old devices. And the economic benefit would be very substantial.
CONFLICT OF INTEREST STATEMENT

Authors state that the manuscript does not have any conflict of interest including any financial, personal or other relationships with other people or organizations within three years of beginning the submitted work that could inappropriately influence, or be perceived to influence, the present work.

ACKNOWLEDGMENT

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REFERENCES


Table 1. Specifications of the gas heating device

<table>
<thead>
<tr>
<th>Parameters</th>
<th>D, t, L (m)</th>
<th>Materials</th>
<th>Quantity</th>
</tr>
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<td>Large cylinder</td>
<td>1.316, 0.008, 2.200</td>
<td>Q235 carbon steel</td>
<td>1</td>
</tr>
<tr>
<td>Fire tube</td>
<td>0.325, 0.008, 2.430</td>
<td>20# carbon steel</td>
<td>1</td>
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<tr>
<td>Flue tube</td>
<td>0.042, 0.003, 2.228</td>
<td>20# carbon steel</td>
<td>24</td>
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<tr>
<td>Convective tube</td>
<td>0.038, 0.003, 7.500</td>
<td>20# carbon steel</td>
<td>9</td>
</tr>
</tbody>
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FIGURE CAPTIONS

Fig. 1. Sketch of a gas heating device

Fig. 2. (a) Sketch of the experimental system and (b) the view at cross-section A-A

Fig. 3. (a) Numerical model simulating a cross-section of the heating device and (b) the enlarged view of the convective tube bundle

Fig. 4. A representative grid system

Fig. 5. Grid-independence test

Fig. 6. Comparison of temperatures simulated and measured at nine locations

Fig. 7. Temperature and flow fields with a single flat guide plate

Fig. 8. Temperature and flow fields with a single arc guide plate

Fig. 9. Temperature and flow fields with double flat guide plates

Fig. 10. The effects of guide plates on the total heat transfer rate and average temperature at the convective tube bundle
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