Gas Condensing Unit with Inner Heat Exchanger

Dagnija Blumberga, Toms Prodanuks, Ivars Veidenbergs, Andra Blumberga

***Abstract*—**Gas condensing units with inner tubes heat exchangers represent third generation technology and differ from second generation heat and mass transfer units, which are fulfilled by passive filling material layer. The first one improves heat and mass transfer by increasing cooled contact surface of gas and condensate drops and film formed in inner tubes heat exchanger. This paper presents a selection of significant factors which influence the heat and mass transfer. Experimental planning is based on the research and analysis of main three independent variables; velocity of water and gas as well as density of spraying. Empirical mathematical models show that the coefficient of heat transfer is used as dependent parameter which depends on two independent variables; water and gas velocity. Empirical model is proved by the use of experimental data of two independent gas condensing units in Lithuania and Russia. Experimental data are processed by the use of heat transfer criteria-Kirpichov number. Results allow drawing the graphical nomogram for the calculation of heat and mass transfer conditions in the innovative and energy efficient gas cooling unit.

***Keywords*—**Gas condensing unit, filling, inner heat exchanger, package, spraying, tunes.

I. Introduction

CURRENTLY, the gas cooling and purification is a significant research area due to several factors. First of all, it is related to energy efficiency improvement and climate change mitigation. Secondly, atmospheric pollution with exhaust fumes causes smog. Therefore, it is necessary to solve gas purification issues.

Gas cooling is influenced by the technological process, gas qualitative and quantitative composition, particulate matter properties and their size, as well as other parameters. One of the most promising equipment for heat recovery is contact condenser in which complex heat and mass transfer processes such as condensation and evaporation take place [1], [2].

Contact condenser structures differ and they are divided into two main groups; indirect contact condensers and direct contact or open condensers. In the indirect contact condenser, heat is transferred between flue gases (water-vapor) and working fluid (cold water) with the help of an enclosing wall. In contrast, the direct contact condenser provides heat transfer between flue gases and working fluid without an enclosing wall [3], [4].

The contact condenser structures mainly differ from gas scrubbers with filling element that is made from tube bundles. The main advantage of this type of apparatus is the inner heat exchanger that intensifies heat and mass transfer processes in contact condenser [5]. It is important to understand the complex heat and mass transfer processes on tube bundles, as well as hydrodynamic fluid film flow on different configuration surfaces if they are exposed to the gas and vapor flow [5], [6].

Flue gas condensation differs from the condensation of pure vapor [7]. Contradictory conclusions can be found about non-condensing gas influence on heat transfer coefficient in the case of vapor condensation. Some researchers claim that a negligible air presence (ε_g = 0.005) in vapor causes heat transfer coefficient reduction by 50% [8]. However, the others state that if ε_g = 0.01 heat transfer is not influenced. Difference in conclusions may be explained by process dependence on geometric characteristics of condensing surface, gas flow patterns and composition, thermo-physical properties, temperature, pressure, non-condensing gas solubility in condensate, as well as by the other factors [9], [10].

For the solution of heat and mass transfer processes in the case of vapor concentration from gas flow, research on local and average heat and mass transfer characteristics, for example, heat transfer coefficient, average temperature of the wall etc. needs to be performed.

II. Methodology

Experimental research on heat and mass transfer in gas condensing unit with filling tubes (Fig. 1) has been conducted to determine heat transfer coefficient values that are affected by various factors. This is necessary to not only understand the heat and mass transfer processes but also to define the condenser efficiency.

The algorithm of heat exchange empirical model development was created. The algorithm contains the main parts to gain positive result. Algorithm is shown in Fig. 1.

The algorithm shown in Fig. 1. includes seven important parts of empirical model creation for gas condenser unit. Research has been conducted in a stationary mode. Heat transfer coefficient has been acquired by using heat that is obtained from exhaust gases by using average logarithmic heat carrier temperature difference. Therefore, dry-bulb temperature and wet-bulb temperature of exhaust gases have been measured in inlet and outlet of the facility, water inlet and outlet, as well as spraying water temperature before nozzles have been determined. Fig. 2 shows the gas condensing unit with inner heat exchanger and its main parts.

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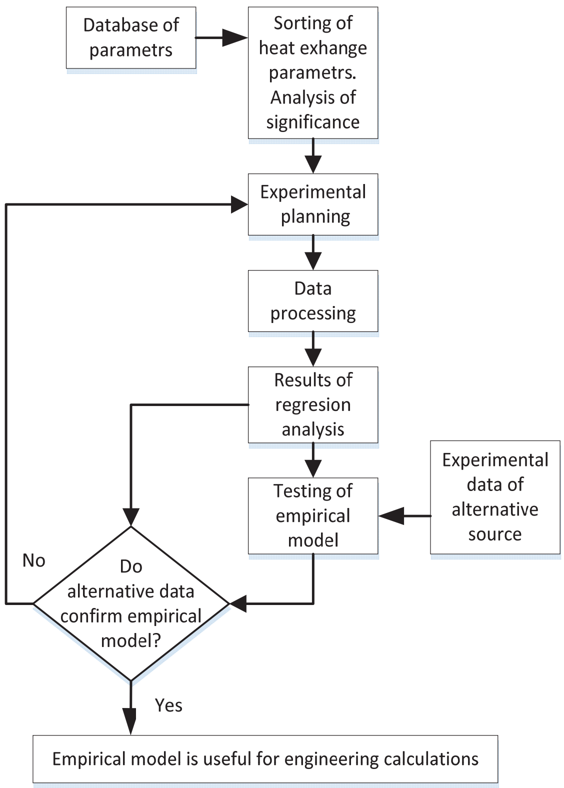

Fig. 1 The algorithm of heat exchange empirical model development

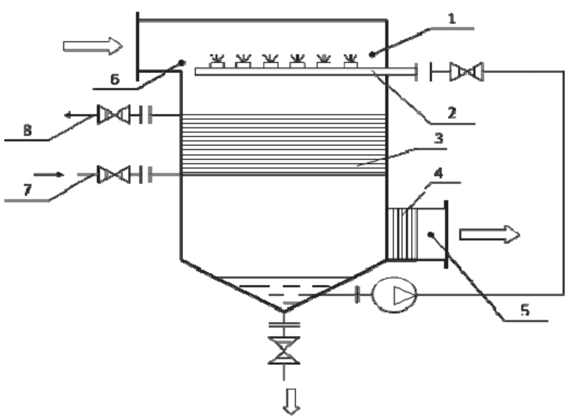

Fig. 2 Gas condenser unit with package tubes: 1 – envelope; 2 – spraying nozzles; 3 – filling tubes; 4 – separation; 5 – gas outlet; 6 – gas inlet; 7 – water inlet; 8 –water outlet

Theoretically, heat transfer coefficient can be calculated using (1):

$$k = \frac{1}{\frac{1}{\alpha_n} + \frac{\delta_{wall}}{\lambda_{wall}} + \frac{1}{\alpha_w}}, \tag{1}$$

where k = heat transfer coefficient (W/m^2·K), α_g = convective heat transfer coefficient of flue gas and water mixture (W/m^2·K), α_w = convective heat transfer coefficient of water inside filling pipes, (W/m^2·K), δ_{wall} = thickness of the filling pipe's wall (m), λ_{wall} = thermal conductivity of filling pipe's wall (W/m·K). However, heat transfer coefficient in condensing units cannot be calculated using (1), because convective heat transfer coefficients are dependent on many variable values, which can only be determined in experiments.

Experimental gas condensing unit has been developed from 1.24 m long filling tubes (outside diameter 20 mm and inside diameter 17 mm). Filling height and width is 0.92 m and 0.54 m, respectively. Heat transfer surface area is 16.4 m^2, but gas flow cross-sectional area in filling is 0.334 m^2.

Modes and their characteristic parameter values have been chosen on the basis of experiment planning. Moreover, facility's operation has been verified in additional modes to evaluate the result of the obtained equation - correspondence of heat transfer coefficient value with the experimentally obtained values.

Pilot study of the facility is based on the three-factor experiment (factorial design) where each factor has been evaluated in two levels: maximum and minimum. Therefore, factorial design of experiment is 2^k, where k = 3, and S is the number of experiments. To evaluate experimental result adequacy plan's center, three studies are made with the average values of factors (all factor values must be in average level).

Chosen factors (independent variables or process parameters) that influence facility's heat exchange processes are flue gas velocity in the smallest cross-sectional area of the filling, water velocity in filling pipes, and spray density. Factor change in facility depends on flue gas flow rate, water flowrate through filling, and spraying device. Experimentally obtained parameter values and results are shown in Table I.

TABLE I
FACTOR REAL VALUES

Trial number	First factor		Second factor		Third factor		Trial result
	V_g	ω_g	G_w	ω_w	G_{lq}	H_{lq}	k
1	1.97	5.9	0.97	0.23	1.35	2.0	174
2	1.97	5.9	3.83	0.78	1.35	2.0	370
3	1.97	5.9	0.97	0.23	4.03	6.0	215
4	1.97	5.9	3.83	0.78	4.03	6.0	443
5	0.49	1.48	0.97	0.23	1.35	2.0	162
6	0.49	1.48	3.83	0.78	1.35	2.0	222
/7	0.49	1.48	0.97	0.23	4.03	6.0	170
8	0.49	1.48	3.83	0.78	4.03	6.0	321
9	1.23	3.7	2.4	0.5	2.69	4.0	260
10	1.23	3.7	2.4	0.5	2.69	4.0	285
11	1.23	3.7	2.4	0.5	2.69	4.0	297

V_g = flue gas capacity (m^3), ω_g = flue gas velocity in the smallest cross-sectional area of the filling (m/s), G_w = water capacity in inner heat exchanger (10^3 m^3/s), ω_w = water velocity in inner heat exchanger pipes (m/s), G_{lq} = water capacity in spraying device (10^3 m^3/s), H_{lq} = spraying density (kg/ m^3·s), k = heat transfer coefficient (W/ m^3·K).

Arithmetic mean value of gas flowrate has been used for experimental data processing. Sprinkling density is ascribed to the cross-section area over the active filling.

During experiment planning and result processing, the chosen hypothesis is that heat transfer coefficient dependence from analyzed factors has a linear tendency. By taking into account the factor interaction double and triple effects, mathematical relationship can be described as linear regression by using (2):

$$y = b_0 + b_1x_1 + b_2x_2 + b_3x_3 + b_{12}x_1x_2 + b_{13}x_1x_3 + b_{23}x_2x_3 + b_{123}x_1x_2x_3, \quad (2)$$

where y = experiment result, b_0 b_1 b_2 b_3 = regression equation coefficients, b_{12} b_{13} b_{23} = factor double interaction coefficients, b_{123} = factor triple interaction coefficients, x_1, x_2, x_3 = first, second and third factor in a dimensionless form.

III. Results

Coefficients in (2) are calculated by using design matrix for full factorial experiment. Conducted regression analysis shows that coefficients b_0, b_1, and b_2 need to be taken into account in regression using (3). Significance of coefficients has been evaluated by taking into account Student t-distribution (t_p) criteria. From the regression coefficients that have not been used in analysis, the next most significant coefficients are b_3 and b_{12}. Coincidence error is S_b = 18. It is evaluated by use of three tests in the center of experiment's plan. By taking into account specific coefficient values, regression (2) is transformed and shown in (3). Equation (3) has been verified by carrying out Fisher test (F-test).

$$y = 259.6 + 40.8x_1 + 79.4x_2 \quad (3)$$

Real parameter values can be mathematically described by (4):

$$k = 46.9 + 18.5\omega_g + 288.7\omega_w \quad (4)$$

To evaluate the accuracy of (3), it is necessary to calculate the measurement error of k (heat transfer coefficient). Heat transfer coefficient, as well as part of the other necessary values for calculations, has been obtained by indirect measurements. Systematic errors of these values are determined with the help of directly measurable value systematic errors. Systematic error of k equals 7.5 W/m^2·K. The repeated measurements show that the heat transfer coefficient value deviation is higher than the systematic error. This is associated with random error in the conducted measurements.

Probable values of random error have been determined from three measurements: mathematical expectation of heat transfer coefficient equals 280.7; mean square error for one separate result is equal to coincidence error S_b = 18.9. If the confidence level is 95%, then confidence interval can be determined by (5). Mean square error and systematic error need to be taken into account in calculation.

$$t_p \cdot \frac{S_n}{\sqrt{m}} + \Delta k = 54.4, \quad (5)$$

where t_p = Student's t – factor, S_n = Mean squared error, m = methodologic coefficient, Δk = systematical error.

Student t-factor value is 4.3, and relative error in the center of experiment's plan equals 19%.

The acquired (4) can be used in calculations if parameter values are within this range: gas flow velocity between 1.3 m/s and 6.5 m/s, water velocity in pipes between 0.2 m/s and 0.8 m/s, spray density between 1.8 kg/m^{2}s and 6.1 kg/m^{2}s.

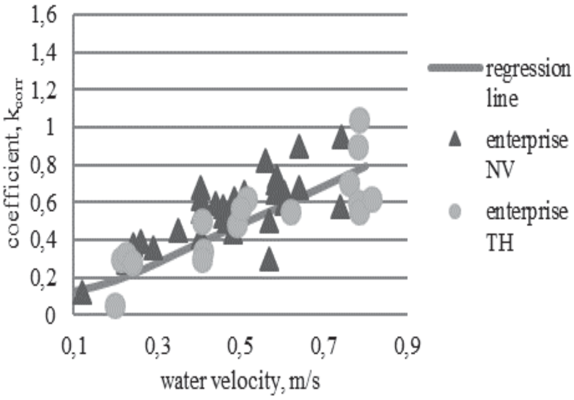

Fig. 3 Regression analysis of heat transfer coefficient

Coefficient (k_{corr}) is expressed from (4), to be able to attribute it to the water velocity (ω_w). Equation (6) shows how k_{corr} is calculated:

$$k_{corr} = \frac{k - 46.9 - 18.5 \cdot \omega_g}{288.7}, \quad (6)$$

Fig. 3 show larger data dispersion, when velocity of water in tubes is increasing. This can be explained by more intensive vapor condensation process. When velocity of water in tubes is increasing, temperature on the tube's wall is decreasing.

Usually experimental results in thermodynamics summarize in criterial equation. To combine the gained experimental data into similarity numbers, it is necessary, from total thermal resistance in heat transfer and from gas mixture for water into filling tubes, which are determined experimentally, to deduct two thermal resistances for: heat dissipation from tubes wall to water and heat dissipation to tubes wall.

The heat dissipation depends on flow rate, temperature, and velocity. Hydrodynamic and thermal stabilization heat dissipation, which is determined with Nusselt number (Nu), decreases and inclines to Nu true class limit, because of boundary layer forming. Reynolds number for water (Re_w) in experimental unit varies from 5000 to 15000, which corresponds to transient and turbulent conditions.

Results show that, in tubes with length (l) of 1.24 m and diameter (d_{in}) of 0.017 m, stabilization of hydrodynamic and thermal processes occur on the significant part of the heat transfer surface. In this condition, heat dissipation coefficient from tubes wall to liquid, which circulates into them can be calculated using (7):

$$Nu_w = 0.012(\mathrm{Re}_w^{0.67} - 280)\,\mathrm{Pr}_w^{0.04}[1 + (\frac{d_{in}}{l})^{\frac{2}{3}}], \quad (7)$$

where Nu_w = Nussel number for water flow inside tubes, Re_w = Reynolds number for water inside tubes, Pr_w = Prandtl number for water inside tubes, d_{in} = tubes inside diameter (m), l = tubes length, m.

Similarity numbers for water in filling determines in average temperature. Equation (7) and filling tubes wall thermal resistivity give the possibility using average parameter values into gas condensing unit to calculate heat transfer coefficient from gas-vapor mixture to tubes outside wall. This value can be summarized using Kirpichov.

$$Ki = \frac{\alpha_g \cdot d_{ou}}{\lambda_g}, \quad (8)$$

where Ki = Kirpichov number, d_{ou} = tubes outside diameter (m), λ_g = thermal conductivity of gas and vapor mixture (W/m·K).

Gas flow conditions in condensation unit can be evaluated using Reynolds number. The heat transfer dependence from gas flow conditions can be defined using (9):

$$Ki = c \cdot \mathrm{Re}_g^{\,m} \cdot \mathrm{Re}_w^{n}, \quad (9)$$

where c = methodologic coefficient, m = methodologic coefficient, n = methodologic coefficient Re_g-Reynolds number for gas flow.

Equation (9) must be turned into logarithmic function (10), which connects Kirpichov number with factor logarithms.

$$\lg Ki = \lg c + m \lg \mathrm{Re}_g + n \lg \mathrm{Re}_w, \quad (10)$$

Using new variables-factor logarithms, full experimental planning and analysis of results were established. In this case, the center of plan corresponds to the average factor logarithm values. That is why, three additional tests are required in the center of the plan.

After processing experimental data, regression equation (11) is developed.

$$\lg Ki = -0.699 + 0.17 \lg \mathrm{Re}_g + 0.59 R_w, \quad (11)$$

Equation (11) can be transferred into (12).

$$Ki = 0.2 R_{eg}^{0.17} R_w^{0.59}, \quad (12)$$

Heat and mass transfer process from the gas and vapor mixture to the filling tubes wall is shown in Fig. 4.

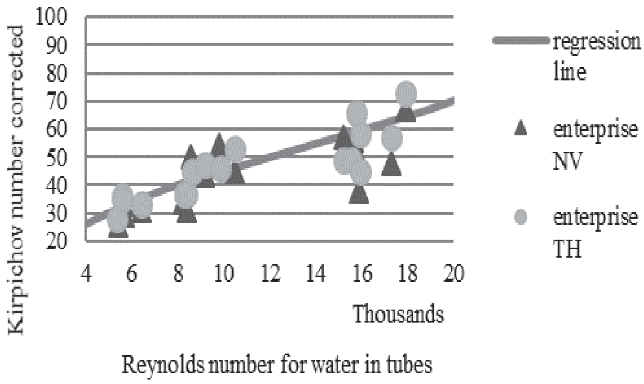

Fig. 4 Regression analysis of corrected Kirpichov heat transfer criteria

Corrected Kirpichov criteria are expressed from (12). Fig. 4 shows acceptable correspondence for empirical model with experimental data from industrial data. That gives opportunity to use these equations for dry heat transfer from gas and vapor mixture to surface of the condensate, thermal resistance of phase change, and liquid film.

IV. Conclusions

The heat exchange research in active filling gas condensers is dependent on 15 to 20 parameters among which three parameters are the most important: water velocity, gas velocity, and sprinkling density.

Mathematical experimental data processing indicates that heat transfer coefficient is dependent on water and gas velocity.

Empirical model (12), which defines Kirpovich similarity criteria is established, to study heat transfer gas through film of condensate to active filling tubes wall.

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