Thermal-hydraulic tests of finned recirculating cooling equipment

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Abstract. The article analyzes the experience of thermal-hydraulic experimental studies of various types of finned tube heat exchangers by the Power Equipment Test Facility of JSC "NPO CKTI". The method of testing the air-cooled heat exchanger with finned heat exchange surface for recirculation cooling units ROPE(A)-2400/253-1 of the Belarusian NPP was developed. The results of the tests according to the developed method are necessary to confirm the thermal characteristics of the heat exchange equipment of the NPP before the start of industrial operation.

1. Introduction

It is required to confirm thermal and hydraulic characteristics of the heat-exchange equipment before exploitation on-site. Such confirmation can be done either in the form of calculation or thermal hydraulic test report. Similarly, a thermal hydraulic test must be performed for heat-exchange equipment of a new construction, i.e. without reference on-site, in order it can be supplied to a Nuclear Power Plant [1, 2]. Therefore, it is currently important to design a universal procedure for testing real heat-exchange equipment which must comprise tests under different conditions such as:

- normal conditions,
- abnormal conditions,

and it should also contain a calculation algorithm of thermal and hydraulic characteristics of the heat-exchange equipment under nominal conditions.

This study describes the procedure for testing the heat-exchange equipment having finned air-cooling surfaces. The procedure is based on results of multiple thermal and hydraulic tests of equipment of various finned types which are performed by NPO CKTI, JSC for more than last 10 years. A detailed description of such test and construction of the equipment can be found in [3, 4]. This study aims to summarize the experience of such tests.

2. Methods

Authors have designed the procedures of thermal and hydraulic tests for a number of finned recirculating cooling unit of the type ROPE(A)-2400/253-1 and project PE 717.00.00.000 used at Belarusian NPP.

The parameters of such equipment in nominal mode are as follows:

- air flow rate Q₂ = (1800…60000) m³/h;
- inlet air temperature T₂,in = (28…60) °C;
- water flow rate G₁ = (0,7…100) t/h;
The test of the recirculating cooling equipment (RCE) has several objectives. First, some designed values, namely heat transfer, outlet airflow pressure, and air mass flow must be equal to modelled values.

3. Test procedure

The test is conducted in Joint-Stock Company “I. I. Polzunov Scientific and Development Association on Research and Design of Power Equipment” center for power equipment. This center is certified by Rosatom. The diagram of the test unit is provided in Figure 1.

![Diagram of a test unit](image)

**Figure 1.** Diagram of a test unit: 1 – slide gate; 2 – exhausting pipe extension; 3 – ventilator; 4 – heat-exchanger; 5 – valves; 6 – regulator valve; 7 – air-water compensator; 8 – electrical heater; 9 – circulation pump; 10 – water stream direction; 10 – air; $P_1$ – ingoing water pressure sensor; $dP_1$ – water differential pressure sensor; $dP_2$ – air differential pressure sensor, $G_1$ – flow meter; $T_{1,in.1,2}$ – ingoing water temperature sensor; $T_{2,in.1,2}$ – outgoing water temperature sensor; $T_{1,out.1,2}$ – inflow air temperature sensor; $T_{2,out.1,2}$ – outgoing air temperature sensor.

Such test can be conducted in an “inversed mode” when the laboratory air is heated by water, which temperature is higher than that of air.

In order to ensure reliability of the test result, it is recommended to repeat the test at least 10 cycles. Furthermore, within each cycle it is important to keep Reynolds numbers for both water and air in nominal mode.

Water flow can be adjusted through changing of pump throughput (7) or opening the valve (4).

When the outgoing air temperature approaches the maximum value $T_{engine}$ for the ventilator’s motor, the air flow should be increased automatically; the water throughput or the temperature of incoming water should be decreased. When the outgoing air temperature will reach the maximum value $T_{engine}$ for the ventilator’s engine, it is required to switch off the fan motor and close the water flow.
The value of water pressure should be maintained at a level, where water boiling in heat-exchanger tubes does not occur. In order to prevent saturation process and taken into account current water temperature, the pressure value of outgoing water should be higher the critical pressure by at least 0.05 MPa.

Due to extensions of pipelines both: the inlet, spreading along 20*DN, and the outlet, spreading along 10*DN, it is possible to stabilize the water and air flow and also estimate friction losses. It is recommended to choose places for pressure sensors and process the experimental data using algorithm in accordance with [5].

The pressure losses the free head at the outlet of the installation (ΔP₂) can be measured by pressure sensors located at the stabilization area at the outlet of the fan. The gate valve is mounted at the end of the stabilizing area. It is possible to measure the value of air head of the unit either from the fan characteristic curve using interpolation or the adjusting the valve’s disc into a position, where the air throughput is nominal.

During the tests are measured:

– air temperature at the inlet of the unit. In order to take into account the temperature stratification of the air in the height of the inlet section of the heat exchanger, it is necessary to use at least two temperature sensors. To avoid distortion of their readings due to radiant heat exchange sensors are located outside the line of sight of the heat exchanger tubes, equidistant from it. One sensor – at the height of the upper (T₂.in.1), and the other – the lower (T₂.in.2) sections of the inlet section of the heat exchanger. The inlet air temperature is defined as

\[ T₂.in = \frac{T₂.in.1 + T₂.in.2 + \ldots + T₂.in.n}{n}; \]

– air temperature at the outlet of the unit. A minimum of two temperature sensors must be used for the measurement. The sensors should be located in the cut-off section of the fan, i.e. in the area where the output air flow is most mixed. The outlet air temperature is defined as

\[ T₂.out = \frac{T₂.out.1 + T₂.out.2 + \ldots + T₂.out.n}{n}; \]

– inlet water temperature (T₁.in.i) into and out of the installation (T₁.out.i). Sensors are installed on stabilization areas covered with thermal insulation;

– inlet water pressure (P₁);

– the differential pressure of the water in the installation, equal to the hydraulic resistance of the heat exchanger installation (dP₁);

– water flow through the installation (G₁);

– the free head at the outlet of the installation (dP₂).

4. Test instruction

Switch on the fan engine. Set the air flow rate, then open water valves. Increasing the power of heater (6), ensure the temperature of ingoing water will reach its minimum value. Collect the experiential data while changing the flow rate of water and air and taking into account the fact that the measurement update speed of sensors’ is fewer than 2% per minute. Then adding power to the heater, increase the temperature of incoming water until it will reach the average value. Collect the experiential data while changing the flow rate of water and air. Finally, increase the temperature of incoming water until it will reach its maximum and collect the experiential data while changing the flow rate of water and air.

5. Experimental data processing

The heat transfer coefficient from water to steam – α₁, W/(m²·K) [6] can be calculated from the equation provided below:

\[
\alpha₁ = \frac{0.023 Pr_e Re_e^{-0.8}}{1 + 2.14 Re_e^{-0.5} (Pr_e^{0.7} - 1)} \cdot \frac{\lambda₁}{d_{in}}, \quad (1)
\]

The rest values, specified in the formula (1), were named in [6].

Thermophysical properties of water must be determined using its average temperature value, which can be computed using following formula: \[T₁.av = 0.5 \cdot (T₁.in + T₁.out).\]
The heat transfer coefficient for air $\alpha_{2,\text{red}}$, W/(m²·K) [6, 7] can be calculated using the formula provided below:

$$\alpha_{2,\text{red}} = \left( \frac{F_{\text{fin}}}{F_{\text{full}}} E \mu_{\text{fin}} \psi_{\text{fin}} + \frac{F_{\text{tub}}}{F_{\text{full}}} \right) \alpha_2$$

(2)

where $\alpha_2 = \frac{N_u \lambda_2}{l_0}$ – the heat transfer coefficient, computed without taking into account heat transfer of fins, W/(m²·K);

$E$ – the fin efficiency coefficient can be calculated using the equation provided below:

$$E = \frac{f_1(\beta r_{\text{fin}})K_1 - f_1(\beta r)K_1(\beta r_{\text{fin}})}{f_1(\beta r)K_1(\beta r_{\text{fin}}) + f_1(\beta r_{\text{fin}})K_0(\beta r)} \cdot \frac{2r}{f_1(r - r')}.$$  

(3)

The rest values, specified in the formula (1), were named in [6, 7].

The tube axial length $l_0$, m in the formula (3) can be calculated using the formula provided below:

$$l_0 = 2(s_{\text{fin}} - \delta_{\text{fin,average}})(s_1 - d_{\text{shint}})/(s_{\text{fin}} - \delta_{\text{fin,average}} + s_1 - d_{\text{shint}}).$$

(4)

$\alpha_2$ is the heat transfer coefficient for circular heat-exchanging tubes with transverse fluted finning, W/(m²·K) can be calculated using the equation below:

$$\alpha_2 = 3.31 R e_2^{0.5} P r_2^{0.33} - 0.78 C_z C_{\alpha} \lambda_2 / d_{\text{hydr}}.$$  

(5)

The calculated value of the heat transfer coefficient $k_{\text{fin}}$ can be estimated from the formula below W/(m²·K):

$$k_{\text{fin}} = \frac{1}{\varphi} \left( \frac{1}{\alpha_{2,\text{red}} \varphi} + \frac{d_{\text{shirt}}}{2 \lambda_{\text{tub}}} \ln \left( \frac{d_{\text{out}}}{d_{\text{in}}} \right) + \frac{d_{\text{shirt}}}{2 \lambda_{\text{gap}}} \ln \left( \frac{d_{\text{out}} + 2 \delta_{\text{gap}}}{d_{\text{out}}} \right) + \frac{d_{\text{shirt}}}{2 \lambda_{\text{fin}}} \ln \left( \frac{d_{\text{shirt}}}{d_{\text{in}} + 2 \delta_{\text{gap}}} \right) + \frac{d_{\text{shirt}}}{\alpha_1 d_{\text{in}}} \right)^{-1}.$$  

(6)

6. Calculation of the friction losses within the tubes of heat-exchanger

Air volume flow $Q_2$ (m³/h) can be determined using the thermodynamic equilibrium equation in relation to water and air:

$$Q_2 = 3600 \frac{N_1}{(i_{2,\text{out}} - i_{2,\text{in}}) \rho_{2,\text{out}}},$$

(7)

where $i_{2,\text{in}}, i_{2,\text{out}}$ are enthalpies of ingoing and outgoing air respectively, J/(kg K);

$\rho_{2,\text{out}}$ is outgoing air density, kg/m³;

$N_1$ is water cooling capacity, Wt, which can be estimated from the formula below:

$$N_1 = G_1 (i_{1,\text{in}} - i_{1,\text{out}}) = N_2,$$

(8)

where $i_{1,\text{in}}, i_{1,\text{out}}$ are enthalpies of ingoing and outgoing water respectively, J/(kg K);

$N_2$ is air cooling capacity, Wt.

After the calculation is complete, it is possible to depict the relation between friction factor ($\zeta_1$) and Reynolds numbers (Re₁) on a chart. Using some analytical methods, such as the least square method, it is possible to compute the friction factor in nominal mode.

Finally, it is possible to determine the overall friction factor ($\Delta P_1$) from the formula below:

$$\Delta P_1 = \zeta_1 \frac{\rho_{1} W_1^2}{2}.$$  

(9)
Conclusion

1. In this article authors summarized the results of thermal-hydraulic experimental studies of various types of heat-exchangers with finned tubes.

2. A method of testing the air-cooled heat exchanger with finned heat exchange surface based on Power Equipment Test Facility of JSC “NPO CKTI” test experience was proposed.

3. This method it currently used for testing new heat-exchange equipment, designed for Nuclear Power Plants.

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