Report on Deterioration of Heat Transfer Performance due to Scale Adherence

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Abstract

An increase in thermal resistance when scale deposits are present is estimated by considering a standard freezing cycle using HFC-134a as an example. The result is generalized, and an expression is derived that can be applied to all freezing systems regardless of the type of refrigerant and the size of the system. The difference in freezing performance (possibility of cost reduction) of the system between when it is operated without scale deposits and when it is operated with scale deposits can be easily estimated from the following formula:

Formula of Freezing Performance Acceleration Ratio due to Scale Removal

$$\frac{Q_{e\,No-scale}}{Q_e} = 1 + \frac{6.65\left(\frac{0.016}{d[\mathrm{m}]}\right)^{0.2}(u[\mathrm{m/s}])^{0.8}\,\delta[\mathrm{mm}]}{1 + 1.87(u[\mathrm{m/s}])^{0.8}}$$

(Oil film on the condensation surface at the refrigerant side is ignored.)

$$\frac{Q_{e\,No-scale}}{Q_e} = 1 + \frac{6.65\left(\frac{0.016}{d[\mathrm{m}]}\right)^{0.2}(u[\mathrm{m/s}])^{0.8}\,\delta[\mathrm{mm}]}{1 + 1.87\left(1 + 0.92\left(\frac{0.016}{d[\mathrm{m}]}\right)^{0.2}\right)(u[\mathrm{m/s}])^{0.8}}$$

(Oil film on the condensation surface at the refrigerant side is considered.)

For example, if the thermal resistance of an oil film on the

condensation surface at the refrigerant side is also considered and if the outer diameter d of a tube of the condenser is 0.016 m, the flow rate u of cooling water is 1 m/s, and the thickness δ of the scale is 0.1 mm, then $Q_{e\ no\ scale}$ / Q_e equals 1.15 according to the above formula. In other words, the operation time of the freezing cycle can be reduced by about 15% by removing the scale deposits so that a cost reduction can be expected.

1. Estimation of Freezing Performance Based on a Standard Freezing Cycle

In order to consider the influence of scale deposits, it is necessary to estimate a heat discharge amount in the condenser during a cooling cycle. The freezing performance Q_e, the required pumping power L, and the heat discharge amount Q_c of the condenser are estimated by considering a standard freezing cycle using HFC-134a as an example. A P-h diagram of HFC-134a is shown below.

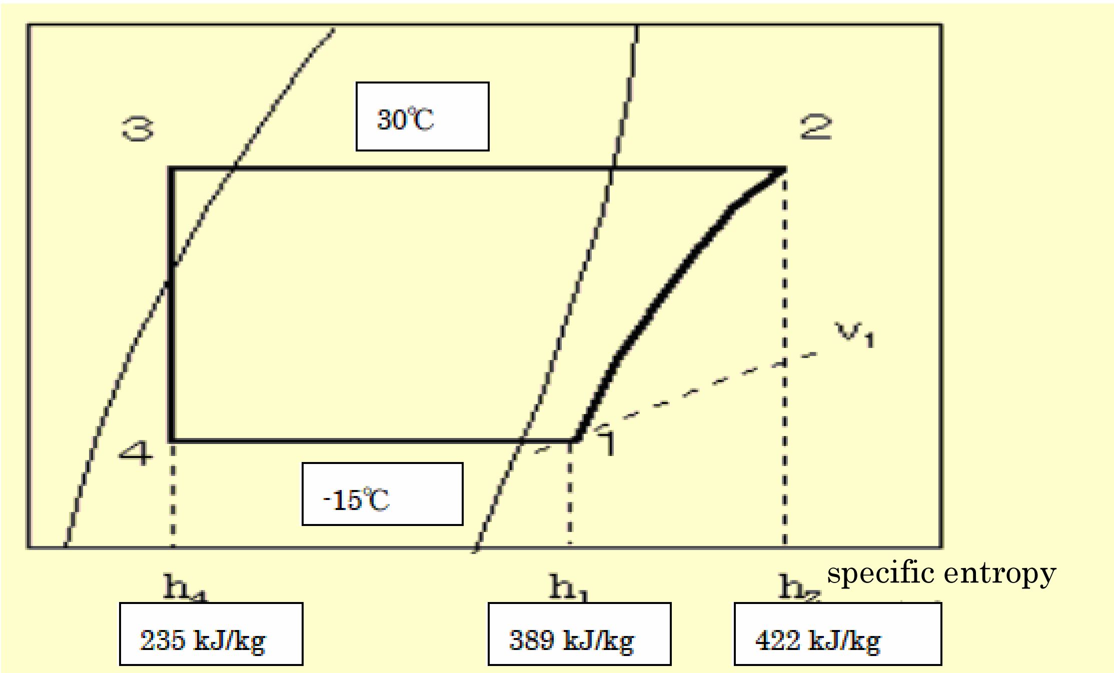

Fig. 1 Standard Freezing Cycle

The freezing performance Q_e, the required pumping power L, and the heat discharge Q_c of a condenser of this standard cycle can be calculated as shown below by reading the specific enthalpy from Fig. 1:

$$Q_e = (h_1 - h_4)G = (389 - 235)(kJ/kg) \times 0.01(kg/s) = 1.54\,kW$$

(1.54 /3.38 = 0.4 tons of refrigerant)

$$L = (h_2 - h_1)G = (422 - 389)(kJ/kg) \times 0.01(kg/s)/\eta_t\eta_{velt} = 0.33\,kW/(0.71 \times 0.96) = 0.48\,kW$$

$$Q_c = (h_2 - h_4)G = (422 - 235)(kJ/kg) \times 0.01(kg/s) = 1.87\,kW$$

The refrigerant circulation flow rate is assumed to be 0.01 kg/s and a typical value is considered for each parameter.

$G = \left(\frac{\pi}{4}D^2 szn\right)\eta_v / v_1 = 0.01\,kg/s$: Refrigerant circulation flow rate

η_r : Total adiabatic compression efficiency

η_{velt} : Velt conduction efficiency

η_v : Volumetric efficiency

v_1 : Specific volume of compressor inlet

$$Q_e / L = 1.54/0.48 = 3.21$$

In other words, 3.21 kW of heat can be absorbed per 1 kW of motive energy. In order to realize this,

$Q_c / L = 1.87/0.48 = 3.90$.

In other words, it is necessary to ensure 3.90 kW of heat discharge per 1 kW of motive energy in the condenser. A serious problem occurs when it cannot be ensured due to scale deposits.

2. Estimation of the Thermal Resistance on the Refrigerant Side of the

Condenser

A general horizontal shell tube condenser is considered.

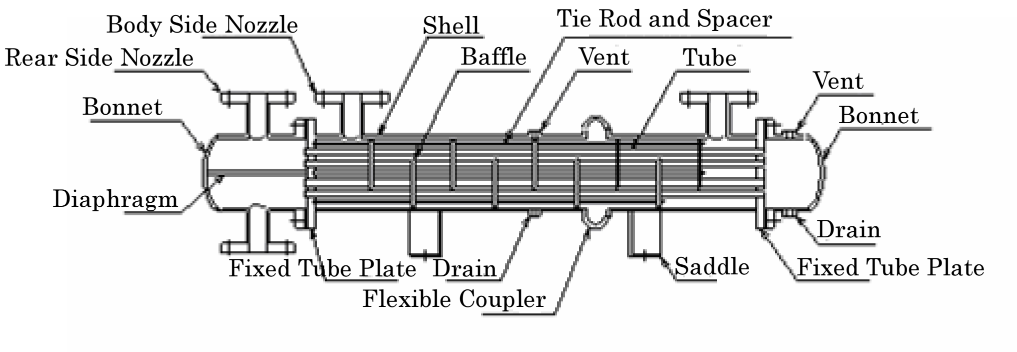

Fig. 2　Horizontal Shell and Tube Condenser

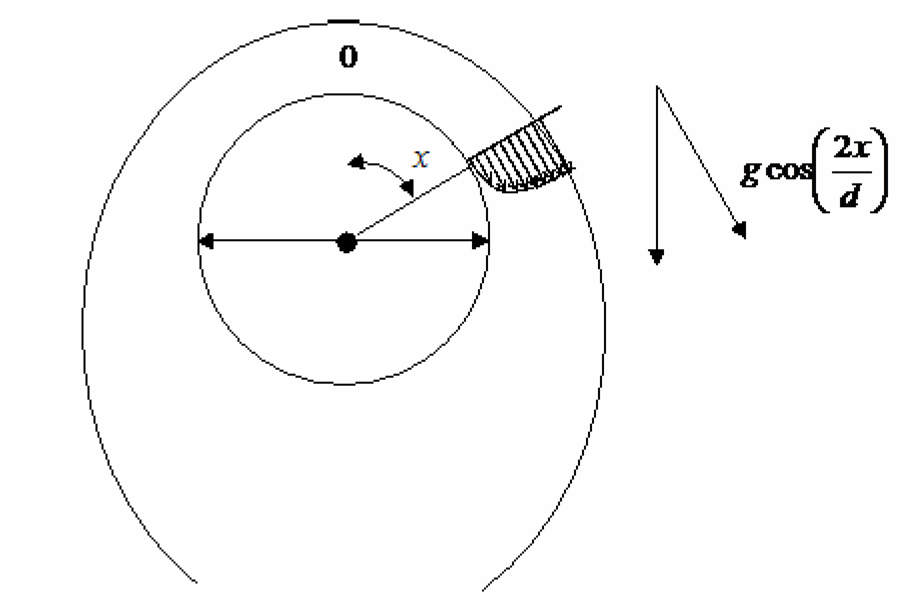

Fig. 3　Condensation Around the Tube

Many copper tubes are horizontally arranged in a copper housing. Cooling water flows into the tubes, and refrigerant vapor around the tubes is continuously condensed. Therefore, the thermal resistance increases when scale deposits adhere to the tubes and there is a risk that the required heat discharge cannot be ensured.

The condensation process is considered by focusing on one of the copper tubes. When an analysis is made by placing a starting point at an upper stagnation point on film-like condensation around a horizontal circular tube having an outer diameter d shown in Fig. 3, the following expression is obtained as a formula for the average heat transfer h_{av} (Thermal Hydraulics, Nakayama, Kuwahara, and Kyo, Kyoritsu Shuppan, 2002, p. 111).

$$\frac{h_{av}d}{k_f} = 0.728\left(\frac{gd^3}{{v_f}^2}\right)^{1/4} {Pr_f}^{1/4}\left(\frac{h_{fg}}{c_{pf}(T_{sat}-T_w)}\right)^{1/4}$$

where

g: Gravitational acceleration 9.8 m/s^2

k_f: Heat transfer coefficient of liquid refrigerant 0.081W/mK

v_f: Coefficient of kinematic viscosity of liquid refrigerant

μ_f / ρ_f = 0.202 x 10^{-3} / 1206 = 0.167 x 10^{-6} m^2/s

c_{pf}: Specific heat at constant pressure of liquid refrigerant 1.43 kJ/kgK

h_{fg}: Latent heat of condensation $h_2 - h_4$ = 187 kJ/kg

Pr_f: Prandtl number of liquid refrigerant

$\mu_f c_{pf} / k_f$ = 0.202 x 10^{-3} x 1430 / 0.081 = 3.57

When the outer diameter d of the tube is 1.6 cm, the temperature T_w of the cooling surface of condensation is 24°C, and the saturation temperature T_{sat} is 30°C, then the heat transfer coefficient (the average heat transfer) of condensation can be estimated as follows:

$$h_{av} = \frac{0.081}{0.016} \times 0.728 \times \left(\frac{9.8 \times 0.016^3}{0.167^2 \times 10^{-12}} \times 3.57 \times \frac{187}{1.43 \times (30-24)} \right)^{0.25} = 2130 \text{ W/m}^2\text{K}$$

In general, 0.8 to 0.9 m^2 of condensation area per one ton of refrigerant is considered to be necessary. Therefore, about 0.4 m^2 of the condensation area may be considered this time. It turns out that the length of the tube is approximately

$$l = \frac{0.4}{12 \times 3.14 \times 0.016} = 0.66\, m$$

when the number n of cooling tubes is 12.

3. Estimation of the Thermal Resistance Before Adherence of Scale Deposits and the Heat Discharge Performance

The heat transfer coefficient h in the tube is calculated using the Dittus-Boelter equation when the inner diameter of the tube is 0.015 m (the thickness of copper is 1 mm) and the flow rate of water is limited to about 1 m/s.

$$\frac{hd_i}{k_w} = 0.023\left(\frac{ud_i}{\nu_w}\right)^{0.8} \mathrm{Pr}_w^{\ 0.3}$$

: (Thermal Hydraulics, Nakayama, Kuwahara, and Kyo, Kyoritsu Shuppan, 2002, p. 84)

$$h = \frac{k_w}{d_i} \times 0.023\left(\frac{ud_i}{\nu_w}\right)^{0.8} \mathrm{Pr}_w^{\ 0.3} = \frac{0.55}{0.015} \times 0.023 \times \left(\frac{1 \times 0.015}{7 \times 10^{-7}}\right)^{0.8} \times 5^{0.3} = 3990\ \mathrm{W/m^2K}$$

where

k_w: Heat transfer coefficient of water 0.55 W/mK

ν_w: Coefficient of kinematic viscosity of water 7 x 10^{-7} m^2/s

d_i: Inner diameter of tube 0.015 m

u: Flow rate of water in tube 1 m/s

Pr_w: Prandtl number of water 5

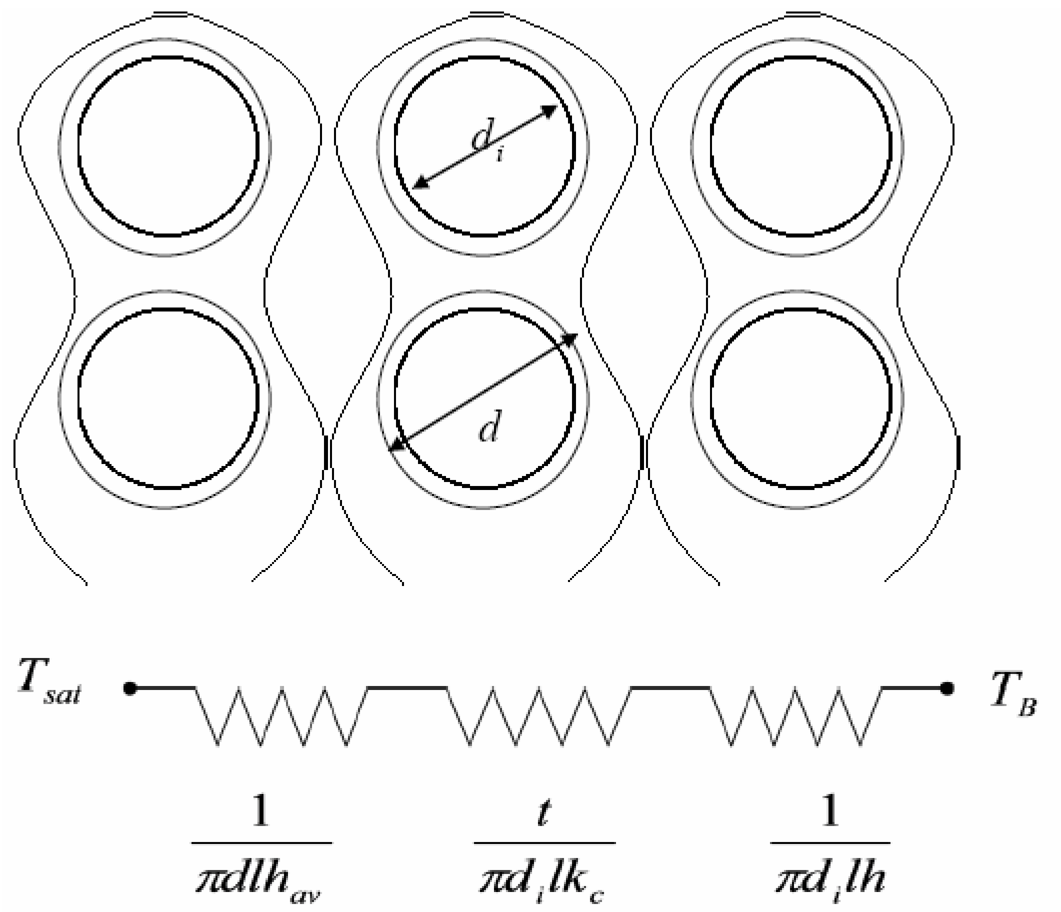

Fig. 4 Thermal Resistance Before Scale Adherence

As shown in Fig. 4, the thermal resistance before scale adherence can be calculated as follows:

$$R = \frac{1}{\pi d l h_{av}} + \frac{t}{\pi d_i l k_c} + \frac{1}{\pi d_i l h} \cong \frac{1}{\pi d l}\left(\frac{1}{h_{av}} + \frac{d - d_i}{2k_c} + \frac{1}{h}\right)$$
$$= \frac{1}{\pi d l}\left(\frac{1}{2130} + \frac{0.001}{2 \times 400} + \frac{1}{3990}\right)$$

where the heat transfer coefficient of copper k_c is 400 W/mK. As found from the above estimation, the thermal resistance of the copper tube is on an order of magnitude that can be ignored as compared with the thermal resistance on the refrigerant side of the condensation surface and that on the water side of the inner wall of the tube, which are equal. Therefore, the thermal resistance before adherence of scale deposits can be approximated with the following formula:

$$R \cong \frac{1}{\pi d l}\left(\frac{1}{h_{av}}+\frac{1}{h}\right)$$

In other words, when the saturation temperature T_{sat} is 30°C and the water temperature of the tube (strictly a logarithmic mean temperature) T_B is 22°C, an estimate can be made with the following formula:

$$n\frac{T_{sat}-T_B}{R} \cong \frac{n(T_{sat}-T_B)}{\frac{1}{\pi d l}\left(\frac{1}{h_{av}}+\frac{1}{h}\right)} = \frac{12\times(30-22)}{\frac{1}{3.14\times 0.016\times 0.66}\left(\frac{1}{2130}+\frac{1}{3990}\right)} = 4.42\,kW$$

This is greater than the required heat discharge Qc (1.87 kW) of the condenser, and it turns out that the condition is satisfied well when no scale deposits are present.

4. Deterioration of the Heat Transfer Performance when Scale Deposits are Present

The total heat resistance is considered as shown in Fig. 5 when the thickness δ of the scales is 0.1 mm and its heat transfer coefficient k_δ is 0.6 W/mK.

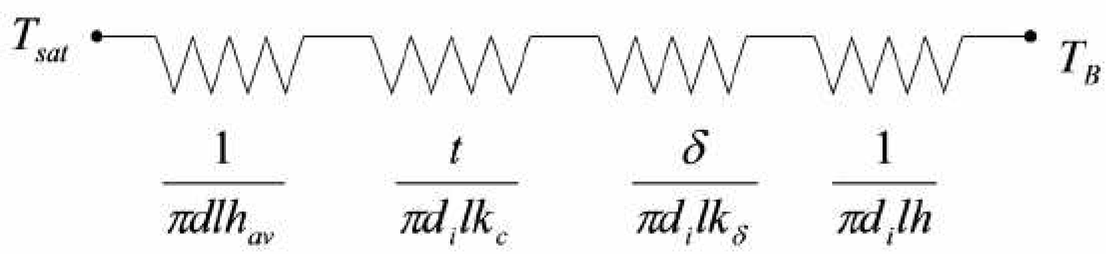

Fig. 5 Heat Resistance when Scale Deposits are Present

$$R = \frac{1}{\pi dlh_{av}} + \frac{t}{\pi d_i lk_c} + \frac{\delta}{\pi d_i lk_\delta} + \frac{1}{\pi d_i lh} \cong \frac{1}{\pi dl}\left(\frac{1}{h_{av}} + \frac{1}{h} + \frac{\delta}{k_\delta}\right)$$
$$= \frac{1}{\pi dl}\left(\frac{1}{2130} + \frac{1}{3990} + \frac{0.0001}{0.6}\right)$$

Therefore,

$$n\frac{T_{sat} - T_B}{R} \cong \frac{n(T_{sat} - T_B)}{\frac{1}{\pi dl}\left(\frac{1}{h_{av}} + \frac{1}{h} + \frac{\delta}{k_\delta}\right)} = \frac{12 \times (30 - 22)}{\frac{1}{3.14 \times 0.016 \times 0.66}\left(\frac{1}{2130} + \frac{1}{3990} + \frac{0.0001}{0.6}\right)} = 3.59\,kW$$

Accordingly, there is a 19% decrease in the cooling ability from 4.42 kW to 3.59 kW due to the scale deposits. Nevertheless, the required heat discharge is attained at this time because a relatively small freezing system is considered, which has the required heat discharge Q_c of the condenser of 1.87 kW, and the thickness of the scales is estimated to be slight (0.1 mm). However, it is a definite possibility that as the scale deposits become thicker, it will no longer be possible to attain the required discharge in a freezing system with a large required heat discharge.

5. Difference Between Freezing Performance During Operation without Scale Deposits and During Operation with Scale Deposits

An increase of thermal resistance when scale deposits are present is estimated by considering a standard freezing cycle using HFC-134a above. The result is generalized, and an expression is derived that can be applied to all freezing systems. With this, the difference between the heat discharge of the system when it is operated without scale deposits and when it is operated with scale deposits can be easily estimated.

It is assumed that the heat discharge $Q_c = (1 + \varepsilon)\, Q_e$ that exactly achieves the freezing performance Q_e is attained during the present operation even though scale deposits are present. In other words,

$$Q_e = \frac{Q_c}{1+\varepsilon} = n\frac{T_{sat} - T_B}{R\,(1+\varepsilon)}$$

When the scale deposits are removed, the heat discharge increases and the freezing performance increases as follows:

$$Q_{e\,No\text{-}scale} = n\frac{T_{sat} - T_B}{R_{No-scale}(1+\varepsilon)}$$

Rationing these values derives the following:

$$\frac{Q_{e\,No-scale}}{Q_e} = \frac{R}{R_{No\text{-}scale}} = \frac{\dfrac{1}{h_{av}} + \dfrac{1}{h} + \dfrac{\delta}{k_\delta}}{\dfrac{1}{h_{av}} + \dfrac{1}{h}} = 1 + \frac{\dfrac{h\delta}{k_\delta}}{1 + \dfrac{h}{h_{av}}}$$

Then, focussing on the proportional connections of $h_{av} \propto d^{-0.25}$ and $h \propto u^{0.8} d^{-0.2}$ yields the following relationship.

$$\frac{h}{h_{av}} = \frac{3990\left(\dfrac{0.016}{d[\mathrm{m}]}\right)^{0.2}\left(\dfrac{u[\mathrm{m/s}]}{1}\right)^{0.8}}{2130\left(\dfrac{0.016}{d[\mathrm{m}]}\right)^{0.25}} = 1.87\left(\frac{d[\mathrm{m}]}{0.016}\right)^{0.05}\left(\frac{u[\mathrm{m/s}]}{1}\right)^{0.8} \cong 1.87(u[\mathrm{m/s}])^{0.8}$$

This is substituted in the above formula, and the result is

$$\frac{Q_{e\,No\text{-}scale}}{Q_e} = 1 + \frac{6.65\left(\frac{0.016}{d[\mathrm{m}]}\right)^{0.2}(u[\mathrm{m/s}])^{0.8}\,\delta[\mathrm{mm}]}{1+1.87(u[\mathrm{m/s}])^{0.8}}$$

Freezing Performance Acceleration Ratio due to Scale Removal

(Oil Film Ignored)

In the above-described example, the outer diameter of a tube of the condenser d = 0.016 m, the flow rate of cooling water u = 1 m/s, and the thickness of scales δ = 0.1 mm, are substituted to obtain:

$$\frac{Q_{e\,No\text{-}scale}}{Q_e} = 1 + \frac{6.65 \times 0.1}{1+1.87} = 1.23$$

.

This is expected to agree with the previous example, 4.42 kW / 3.59 kW = 1.23. In other words, about 23% of the operating time of the freezing cycle can be reduced by removing the scales. Therefore, a cost reduction can be expected.

The heat resistance due to the adherence of oil film to the condensation surface on the refrigerant side has been ignored for the sake of simplicity here. However, there are many cases where it cannot be ignored. A typical example is a case in which the thickness of the oil film is 0.05 mm, and the heat transfer coefficient is approximately 0.116 W/mK (Design and Drafting of Refrigerators, Hasumi, Rikogakusha Publishing Co., Ltd., 1994, p. 43). Because the heat resistance becomes 0.00005 / 0.116 = 0.00043 km^2/W in this case, there is a possibility that the heat resistance will be the same order of magnitude as δ / k_δ = 0.0001 / 0.6 = 0.00017 km² / W with the scale deposits. Taking this into consideration, we obtain:

$$\frac{Q_{e\,No\text{-}scale}}{Q_e} = \frac{R}{R_{No\text{-}scale}} = \frac{\dfrac{1}{h_{av}} + \dfrac{1}{h} + \dfrac{\delta}{k_\delta} + 0.00043}{\dfrac{1}{h_{av}} + \dfrac{1}{h} + 0.00043}.$$

In other words, when the heat resistance is affected by an oil film on the condensation surface of the tube on the refrigerant side, the freezing performance can be estimated as follows.

$$\boxed{\frac{Q_{e\,No\text{-}scale}}{Q_e} = 1 + \frac{6.65\left(\dfrac{0.016}{d[\mathrm{m}]}\right)^{0.2}(u[\mathrm{m/s}])^{0.8}\,\delta[\mathrm{mm}]}{1 + 1.87\left(1 + 0.92\left(\dfrac{0.016}{d[\mathrm{m}]}\right)^{0.2}\right)(u[\mathrm{m/s}])^{0.8}}}$$

Freezing Performance Acceleration Ratio due to Scale Removal

(Oil Film Considered)

In the above-described example, the outer diameter of tube of the condenser d = 0.016 m, the flow rate of cooling water u = 1 m/s, and the thickness of scales δ = 0.1 mm, are substituted to obtain:

$$\frac{Q_{e\,No\text{-}scale}}{Q_e} = 1 + \frac{0.665}{1 + 1.87(1 + 0.92)} = 1.15$$

,

and a 15% cost reduction can be expected.