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Capacity evaluation for refrigerating unit coolers

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Foreword

This Japanese Industrial Standard has been revised by the Minister of Economy, Trade and Industry through deliberations at the Japanese Industrial Standards Committee as the result of proposal for revision of Japanese Industrial Standard submitted by The Japan Cooling Machines and Tools Association (JCA)/Japanese Standards Association (JSA) with a draft being attached, based on the provision of Article 12, paragraph (1) of the Industrial Standardization Act applied mutatis mutandis pursuant to the provision of Article 16 of the said Act. This edition replaces the previous edition (**JIS B 8610**:2002), which has been technically revised.

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Capacity evaluation for refrigerating unit coolers

1 Scope

This Japanese Industrial Standard specifies the details of capacity evaluation for direct expansion refrigerating unit coolers (hereafter referred to as unit coolers).

It is applicable to unit coolers using R410A and R404A refrigerants.

NOTE 1 This Standard provides specific figures and calculation methods for unit coolers using copper pipes of 9.52 mm, 12.70 mm and 15.88 mm in diameter that are fitted with smooth aluminium plate fins. Provided the manufacturer prepares the necessary data by testing, this Standard is also applicable to unit coolers that use other pipe sizes and are fitted with smooth aluminium plate fins or formed aluminium fins.

NOTE 2 The scope of this Standard is limited to "refrigerating" unit coolers so as to exclude air conditioning unit coolers with an evaporating temperature 0 °C or higher which, if included, would cause variation of performance according to the temperature and humidity conditions at the unit cooler inlet, and complicate the calculation formula and diagrams.

The heat exchanging part of a unit cooler is considered to be a dry heating surface without frosting.

2 Terms and definitions

For the purpose of this Standard, the following terms and definitions apply.

2.1

refrigerating capacity

result of multiplying the logarithmic mean temperature difference between the refrigerant evaporating temperature in pipe and the air temperatures at the inlet and outlet of the heat exchanging part of a unit cooler by the overall heat transfer coefficient and the effective external heating surface area [see Formula (1) given in Clause 5]

2.2

external heating surface area

total heating surface area on the ventilating side of a unit cooler [see Formula (4) given in Clause 5]

2.3

effective external heating surface area

effective heating surface area on the ventilating side of a unit cooler that results from the external heating surface area corrected by the fin efficiency and the fin contact efficiency

2.4

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effective internal heating surface area

internal surface area of a pipe excluding bends

2.5

overall heat transfer coefficient

coefficient of heat quantity transferred uniformly through the external heating surface area for a temperature difference of 1 K

2.6

length of superheated portion

length of a portion, per one circuit of pipe, where the refrigerant in the pipe has completely evaporated and the refrigerant vapour is superheated

2.7

effective heating area ratio

ratio of effective external heating surface area to the external heating surface area

2.8

effective internal and external area ratio

ratio of the effective external heating surface area to the effective internal heating surface area

2.9

number of pipe rows

number of rows of pipes in the direction of air flow

2.10

number of pipe stages

number of stages of pipes in the direction perpendicular to the air flow

2.11

effective pipe length

distance between the side plates of the unit cooler

2.12

pipe pitch

pitches of pipe rows and pipe stages (see Figure 1)

2.13

length of fin collar, collar height

value obtained by subtracting the fin thickness from the total length of the copper pipe insertion portion of the fin (see Figure 1)

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2.14

fin efficiency

ratio of the actual heat transfer amount at fin surfaces to that between air and fins, when the temperature of the total fin surface is assumed to be uniformly equal to the temperature at roots of fins

2.15

contact efficiency

ratio of the fin efficiency with thermal resistance existing in contact parts of pipes and fin collars to that without such thermal resistance

2.16

logarithmic mean temperature difference

logarithmic mean of temperature difference between the air temperatures at the inlet and outlet of the heat exchanging part of a unit cooler and the refrigerant evaporating temperature in pipe [see Formula (2) given in Clause **5**]

2.17

temperature difference

difference between the air temperature at the inlet of the heat exchanging part of a unit cooler (room temperature) and the refrigerant evaporating temperature in pipe [see Formula (C.2) given in Annex C]

2.18

front air velocity

mean air velocity at the inlet of the heat exchanging part of a unit cooler

3 Symbols and units

For the purpose this Standard, the symbols and units given in Table 1 apply.

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Table 1　Symbols and units

Symbol	Definition	Unit
Φ_0	Refrigerating capacity	W
A	External heating surface area	m^2
A_e	Effective external heating surface area	m^2
A_i	Effective internal heating surface area (excluding bends)	m^2
A_P	External surface area of pipe	m^2
A_f	Fin one-side surface area (excluding the area of pipe hole)	m^2
A_{af}	Frontal area of unit cooler	m^2
K	Overall heat transfer coefficient	$W/(m^2 \cdot K)$
a	Length of superheated portion (per pipe circuit)	m
d_0	Outside diameter (diameter) of pipe	m
d_i	Inside diameter of pipe	m
k	Effective heating area ratio (A_e/A)	—
l	Pipe length per circuit	m
l_t	Total length of pipe (excluding bends)	m
N	Number of pipes	—
m	Effective internal and external area ratio (A_e/A_i)	—
n_r	Number of pipe rows	—
n_s	Number of pipe stages	—
E_L	Effective pipe length	mm
n	Number of fins	—
f_p [a)]	Fin pitch	mm
P_p [a)]	Pipe pitch	mm
t_c [a)]	Length of fin collar (collar height)	mm
t_f [a)]	Fin thickness	mm
η_f	Fin efficiency	—
η_c	Contact efficiency	—
t_{a1}	Air temperature at the inlet of unit cooler (room temperature)	°C
t_{a2}	Air temperature at the outlet of unit cooler	°C
t_0	Refrigerant evaporating temperature in pipe	°C
t_k	Condensation temperature	°C
α_a	Heat transfer coefficient at the air side of unit cooler	$W/(m^2 \cdot K)$
α_r	Heat transfer coefficient at the refrigerant side	$W/(m^2 \cdot K)$
α_{rG}	Heat transfer coefficient of superheated gas of refrigerant	$W/(m^2 \cdot K)$
φ_a	Heat flow density at the air side of unit cooler	W/m^2
φ_{af}	Refrigerating capacity of unit cooler per unit frontal area	W/m^2
φ_r	Heat flow density at the refrigerant side of unit cooler	W/m^2
Δh	Refrigerating effect (enthalpy difference)	kJ/kg

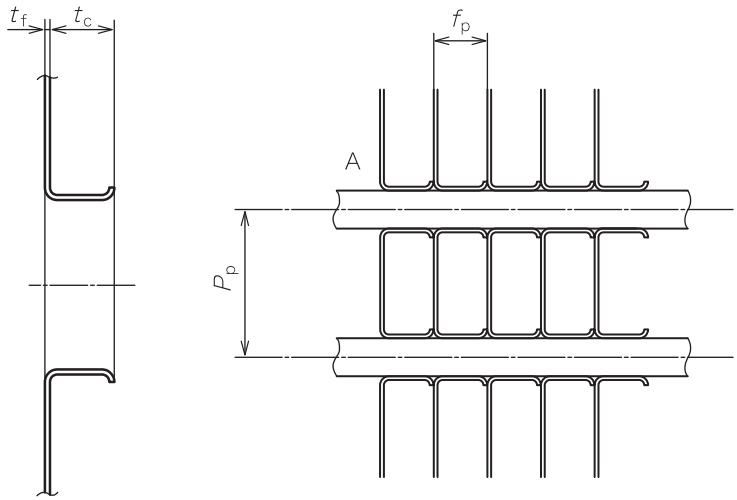
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Table 1 (concluded)

Symbol	Definition	Unit
Δh_{sup}	Refrigerating effect of superheated refrigerant gas at the evaporator outlet and dry saturated vapour (enthalpy difference)	kJ/kg
Δt_{sup}	Degree of superheating of refrigerant vapour at the pipe outlet	K
$\overline{\Delta t}$	Logarithmic mean temperature difference	K
TD	Temperature difference ($t_{a1} - t_0$)	K
$\overline{\Delta t}_{sup}$	Mean temperature difference of superheated portion	K
U_{af}	Front air velocity	m/s
q_{mr}	Refrigerant flow rate in one pipe circuit	kg/h
Δp	Pressure loss	MPa
N_{uG}	Nusselt number of refrigerant gas	—
R_{eG}	Reynolds number of refrigerant gas	—
P_{rG}	Prandtl number of refrigerant gas	—
λ_{rG}	Heat transfer coefficient of refrigerant gas	W/(m·K)
T	Absolute temperature	K
t_R	Room temperature	°C
ρ_a	Density of air	kg/m^3
x	Dryness	kg/kg
μ_L	Viscosity of refrigerant liquid	Pa·s
λ_L	Heat transfer coefficient of refrigerant liquid	W/(m·K)
R_e	Reynolds number at the refrigerant side	—
g	Acceleration of gravity	9.81 m/s^2
c_p	Constant pressure specific heat	kJ/(kg·K)
Note a) f_p, P_p, t_c and t_f shall be as shown in Figure 1.		



Detailed view of A

Figure 1 Enlarged view of fin collar, pipe pitch and fin pitch

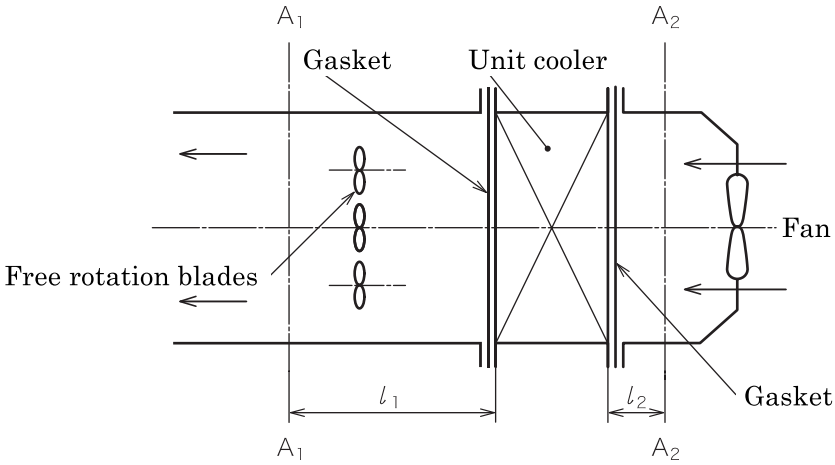
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4 Measurement of air velocity

4.1 Measurement of front air velocity used for calculation of refrigerating capacity

The air velocity shall be measured by means of a hot wire anemometer, a Pitot tube with a micromanometer, or a Biram's anemometer at the section A_1 – A_1 shown in Figure 2 if the unit cooler is provided with a fan at the air inlet side, and at the section A_2 – A_2 at the air inlet side if it is provided with a fan at the air outlet side.

Measurement shall be made at four positions for the air passage cross-section of up to 0.16 m², adding one or more positions for every 0.16 m² increase in air passage cross-section. For this measurement, the air velocity inside a duct at a measuring position shall be made uniform using free rotation blades etc.



Unit: mm

Outside diameter of pipe	l_1	l_2
9.52	200 min.	60 min.
12.70	270 min.	80 min.
15.88	330 min.	100 min.

Figure 2 Measurement of air velocity

4.2 Accuracy of anemometer

The anemometer used for measurement shall be accurate to ±0.2 m/s.

5 Refrigerating capacity

The refrigerating capacity, Φ_0, shall be calculated according to Formula (1) :

$$\Phi_0 = k \cdot A \cdot \left(\frac{l-a}{l} \right) \cdot K \cdot \overline{\Delta t} \quad \cdots\cdots (1)$$

where,

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$$\overline{\Delta t} = \frac{t_{a1} - t_{a2}}{\ln \dfrac{t_{a1} - t_0}{t_{a2} - t_0}} \quad \cdots\cdots (2)$$

$$k = A_e / A \quad \cdots\cdots (3)$$

$$A = \left(A_p - \pi d_0 N n t_f \times 10^{-3}\right) + 2\, n A_f \quad \cdots\cdots (4)$$

$$A_e = \left[A_p - \pi d_0 N n \left(t_f + t_c\right) \times 10^{-3}\right] + \eta_c \left(\pi d_0 N n t_c \times 10^{-3} + 2 \eta_f n A_f\right) \quad \cdots\cdots (5)$$

The k-value in the case of η_c = 1 is shown in Figures 6 to 8, the K-value, in Figures 9 to 11, and the value a, in Figures 12 to 14.

The numerical values in the figures are reference values for calculation.

6 Factors related to performance calculation

6.1 Fin efficiency

The heating surface area of a unit cooler can be used as an indirect index of the refrigerating capacity of the unit cooler. However, the effective external heating surface area, which contributes to cooling action, is affected by the fin efficiency and the contact efficiency between the fin and the coil, and the fin efficiency, by the front air velocity, pipe pitch, and type and thickness of fin material. Figures 3 to 5 illustrate the relationship between the fin efficiency and pipe pitch of 9.52 mm-, 12.70 mm- and 15.88 mm-diameter copper pipe unit coolers with continuous fins.

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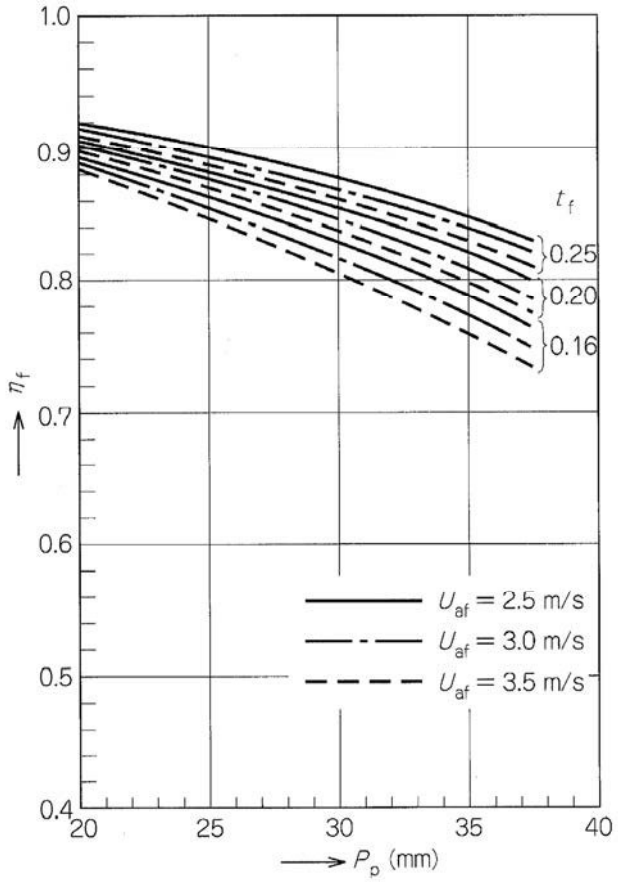


Figure 3 Relationship between fin efficiency, η_f, and pipe pitch, P_P, of 9.52 mm-diameter copper pipe unit coolers with continuous fins

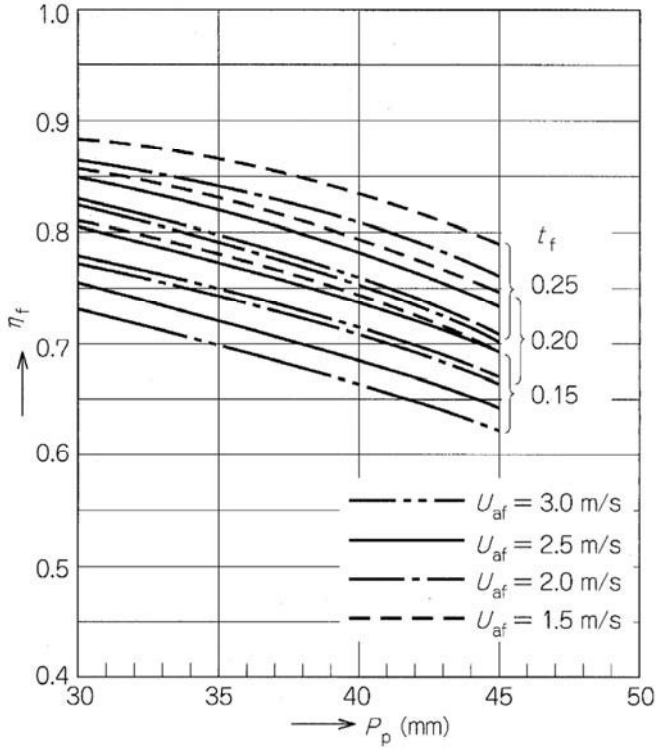


Figure 4 Relationship between fin efficiency, η_f, and pipe pitch, P_P, of 12.70 mm-diameter copper pipe unit coolers with continuous fins

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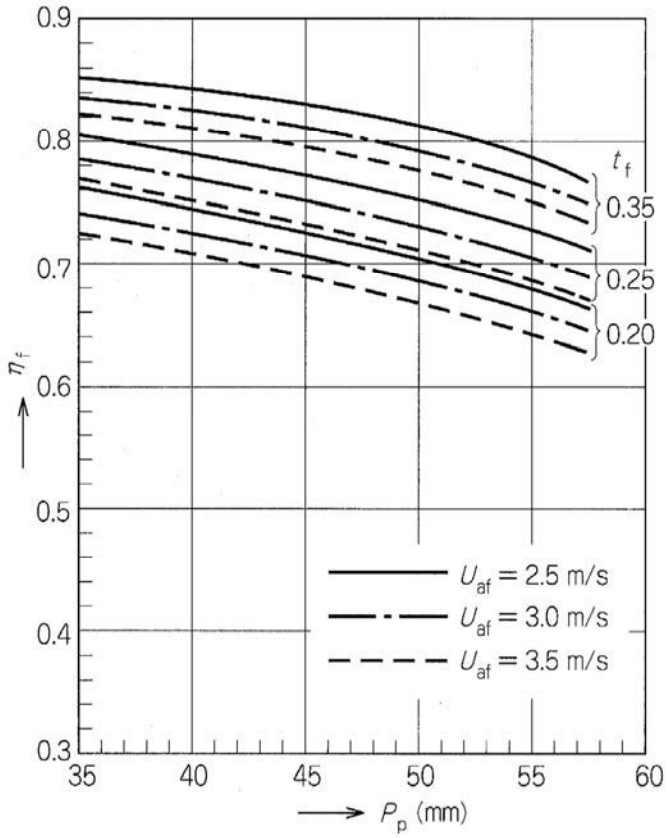


Figure 5 Relationship between fin efficiency, η_f, and pipe pitch, P_P, of 15.88 mm-diameter copper pipe unit coolers with continuous fins

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6.2 Effective heating surface area

To calculate the overall heat transfer coefficient, K (hereafter referred to as K-value) of a unit cooler specified in **6.4**, the effective heating surface area needs to be determined. For this, a "k" value is necessary which is used for obtaining the actually effective heating surface area from the heating surface area at the air side of a product. The "k" values when the contact efficiency $\eta_c = 1$ are shown in Figures 6 to 8. This calculation is based on η_f shown in Figures 3 to 5. When $\eta_c < 1$, the "k" value shall be calculated using Formulae (4) and (5) and Figures 3 to 5. When $\eta_c > 0.97$, as the error of refrigerating capacity is expected to be within 3 %, the "k" value in Figures 6 to 8 may be used.

Since, as shown in **6.4**, the quantity of heat exchanged with the air at the refrigerant gas superheated portion near the unit cooler outlet can be ignored due to being very small, the effective external heating surface area where cooling takes place, as shown in Formula (1), is as follows

$$k \cdot A \cdot \left(\frac{l - a}{l} \right)$$

The superheated portion shall not be taken into consideration for flooded type or forced circulation type unit coolers, where heat transfer is effectively performed with the liquid refrigerant filled up to the coil ends.

The length of superheated portion, a, (hereafter referred to as the a-value) that is used for this formula and shown in Figures 12 to 14 is calculated by the following formulae :

$$m\pi d_i a K \overline{\Delta t}_{sup} = q_{mr} \Delta h_{sup} / 3\,600 \quad \cdots\cdots (6)$$

$$\frac{1}{K} = \frac{1}{\alpha_a} + \frac{m}{\alpha_{rG}} \quad \cdots\cdots (7)$$

$$N_{uG} = \frac{\alpha_{rG} d_i}{\lambda_{rG}} = 0.023\, R_{eG}^{\ 0.8} \cdot P_{rG}^{\ 0.4} \quad \cdots\cdots (8)$$

$$\alpha_a = 40.7 \cdot U_{af}^{\ 0.47} \quad (9.52\ \text{mm-diameter copper pipe } P_p = 22 \times 25.4) \quad \cdots\cdots (9)$$

$$\alpha_a = 30.5 \cdot U_{af}^{\ 0.65} \quad (12.70\ \text{mm-diameter copper pipe } P_p = 33 \times 38) \quad \cdots\cdots (10)$$

$$\alpha_a = 30.2 \cdot U_{af}^{\ 0.63} \quad (15.88\ \text{mm-diameter copper pipe } P_p = 40 \times 40) \quad \cdots\cdots (11)$$

The refrigerant flow rate in one pipe circuit, q_{mr}, used in Formula (6) is a value at completion of evaporation within each pipe length, and determined according to **6.4** respectively for the front air velocity of the coil, U_{af}, effective internal and external area ratio, m, and logarithmic mean temperature difference, $\overline{\Delta t}$ and other conditions, taking as the basic length the pipe length per circuit of 10 m for a 9.52 mm-diameter copper pipe, that of 12.5 m for a 12.70 mm-diameter copper pipe or that of 15 m for a 15.88 mm-diameter copper pipe. Consequently, the pipe length per circuit added with a-value is greater than the above basic lengths.

For q_{mr} thus obtained, the enthalpy increment for the specified degree of superheat-

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ing, Δh_{sup}, shall be determined based on the *p-h* characteristics, and the pipe length required for superheating, *a*, shall be determined using Formula (6) and the *K*-value of the superheated portion obtained by Formulae (7) to (11).

6.3 Mean temperature difference of superheated portion, $\overline{\Delta t}_{sup}$

The *a*-value per one pipe circuit for obtaining the same degree of superheating depends on the supply method of refrigerant and cooling air to the unit cooler. $\overline{\Delta t}_{sup}$ for calculating *a*-value shown in Figures 12 to 14 is obtained according to Formula (12) :

$$\overline{\Delta t}_{sup} = \overline{\Delta t} - \frac{\Delta t_{sup}}{2} \quad \cdots\cdots (12)$$

Since in an actual product, the coil length required for superheating, i.e. *a*-value, may be smaller or greater than the value determined by Formula (12), *a*-value that takes account of the mean temperature difference, $\overline{\Delta t}_{sup}$, between blown air and superheated refrigerant gas of that product should be determined and used for calculation of Φ_0 according to Formula (1).

The value of degree of superheating used in Formula (12) shall be an arithmetic mean.

6.4 Overall heat transfer coefficient, *K*

In a dry direct expansion type unit cooler, as it uses an automatic thermal expansion valve, the refrigerant gas at the outlet side of the unit cooler is in the superheated state. This results in a fairly small heat transfer coefficient at the refrigerant side in this area, making the *K*-value smaller than that of the heat exchanging part in the case of an air-cooled type. Furthermore, since *a*-value is generally equal to 10 % to 20 % of the pipe length per circuit, whether the heating surface area used in the *K*-value calculation includes this value or not may significantly affect the calculation result of the refrigerating capacity.

The *K*-value to be used shall be determined from the effective external heating surface area, A_e, and the logarithmic mean temperature difference, $\overline{\Delta t}$, of the unit cooler. When the influence of the lubricating oil is not considered and the refrigerant in the coil evaporates completely within a given pipe length, *l*, the *K*-value for the effective external heating surface area of a unit cooler using R410A refrigerant can be obtained according to Formulae (13) to (15) :

$$\varphi_r \pi d_i l = q_{mr} \frac{\Delta h}{3.6} \quad \cdots\cdots (13)$$

$$\alpha_r = 0.01 \left(R_e{}^2 \cdot \frac{1\,000 \Delta h}{lg} \right)^{0.4} \cdot \frac{\lambda_L}{d_i} \quad \cdots\cdots (14)$$

$$R_e = \frac{4\, q_{mr} d_i}{3\,600\, \pi d_i{}^2 \mu_L} = \frac{4\, q_{mr}}{3\,600\, \pi d_i \mu_L}$$

$$\frac{1}{K} = \frac{1}{\alpha_a} + \frac{m}{\alpha_r} \quad \cdots\cdots (15)$$

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