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Heat transfer property of refrigerant-oil mixture in a flooded evaporator: The role of bubble formation and oil retention

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Abstract−We examined the effect of oil retention on the heat transfer performance of a shell-and-tube-type evaporator which had 26 inner tubes and was filled with the refrigerant R-134a. The refrigerant was boiled on the surface of the inner tubes in the evaporator, while chilled water circulated through these tubes. An experimental apparatus was designed to measure both the pressure and temperature profiles at the inlet and outlet of the flooded evaporator. Four windows were installed for observing the operation of the flooded evaporator. A series of experiments were carried out under the following conditions: the refrigerant saturation temperature, 5 °C; refrigerant inlet quality, 0.1; heat fluxes from water to the refrigerant, 5-7 kW/m^2. The concentration of the oil retained in the refrigerant was then varied up to approximately 10% to observe the effect on the heat transfer performance of the flooded evaporator. Increasing the oil content (i.e., increasing the concentration up to a maximum of approximately 10%) in the refrigerant R134a did not lead to any appreciable reduction in the overall heat transfer coefficient of a flooded evaporator with multiple-inner-tubes. When the oil concentration in the refrigerant was approximately 10%, the heat transfer degradation in the case of the flooded evaporator with multiple-inner-tubes was approximately 11%, which was found to be much smaller than the heat transfer degradation in the case of a flooded evaporator with a single-tube (26-49%). This observation suggested that the oil retained in the refrigerant did not significantly deteriorate the heat transfer performance of the flooded evaporator, presumably because the presence of tube bundles promoted forced convection by agitating bubbles.

Key words: Flooded Evaporator, Shell-and-tube-type Heat Exchanger, R-134a/Oil Mixture, Enhanced Tube, Heat Transfer Degradation

INTRODUCTION

The refrigeration systems for industrial use should not only be able to withstand high cooling loads but also be robust to large variations in cooling loads. It is known that turbo refrigeration systems with a flooded evaporator are capable of dealing with large amounts of refrigerants; moreover, they can be effectively operated under partial loads, and hence are widely used in industrial refrigeration systems. Generally, in turbo refrigeration systems, shell-and-tube-type heat exchangers are used as evaporators; in these exchangers, chilled water flows inside tubes and the refrigerant evaporates on the outer surface of the tubes [1,2]. Vapor compression refrigeration systems generally employ oil-lubricated compressors, and a mixture of oil and refrigerant is circulated in these systems. Here, the concentration of oil in the refrigerant can reach up to 5% [3]. It is interesting to note that the oil concentration in the refrigerant affects the performance of a flooded refrigerant evaporator. The heat transfer coefficient of the shell side of the flooded refrigerant evaporator can be calculated by adding the nucleate boiling heat transfer coefficient and the forced convective heat transfer coefficient, as suggested by Chen [4]. Therefore, a knowledge of nucleate pool boiling heat transfer coefficient of tubes is essential for predicting the evaporator performance.

The effects of oil concentration on the pool boiling of refrigerant-oil mixtures in structured surfaces have been examined in several previous studies. In general, the addition of oil to boiling refrigerants significantly reduces the heat transfer coefficient of the refrigerant. This reduction occurs because an oil-rich layer is formed on the surface of the tubes owing to refrigerant evaporation; the formation of this layer results in the additional resistance to the boiling heat transfer. The reduction in the heat transfer coefficient is expected to be much more significant in the case of enhanced tubes than in the case of smooth tubes.

Kim et al. [5] tested a single tube that contained R-123 and was operated at 4.4 °C and 26.7 °C and an oil concentration between 0 and 10%. When the oil concentration was increased, significant heat transfer degradation was observed for Turbo-B-type tubes. At an oil concentration of 5%, the heat transfer degradation was 26-49% and 50-67% at T_{sat} values of 4.4 °C and 26.7 °C, respectively. The heat transfer degradation can be significant when a small amount of oil is present in the refrigerant. This is probably because of the accumulation of oil in sub-tunnels. Webb et al. [6] performed pool-boiling tests for a single tube immersed in the mixtures of R-11, R-123, and oil. The concentration of oil in the refrigerants R-11 and R-123 was varied; in particular, the concentration was 0, 0.5, 1, 2,

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and 5%, respectively. For both the refrigerants, an increase in the oil concentration degraded the boiling performance of tubes. This degradation was more in the case of enhanced tubes than in the case of plain tubes. The heat transfer performance of the enhanced tubes was decreased by approximately 10% in the case of R-123 and by approximately 20% in the case of R-11 for an oil concentration of 2%.

Zheng [7] and Chyu [8] tested a 3×5 enhanced tube bundle with an ammonia/lubricant mixture with oil concentration between 0 and 10%. The heat transfer coefficient of enhanced tube bundle was higher than that of single-tube due to two-phase flow convection. However, because of the lubricant, the heat transfer coefficient was lower in the lower rows and higher in the higher rows; therefore, there was a larger variation in the data.

In this study, we examined the effect of oil retention in a refrigerant on the heat transfer performance of a flooded evaporator with multiple-inner-tubes, which is a shell-and-tube-type heat exchanger with Turbo-E tubes.

EXPERIMENTAL

Fig. 1 is a schematic of our experimental setup. We aimed to investigate the effects of oil concentration on the overall heat transfer rate during the evaporation of R-134a. The setup consisted of a refrigerant loop, a water loop, and a water-glycol loop. In the refrigerant loop, the refrigerant was pumped from the receiver and subsequently supplied to the pre-heater, where it was evaporated. This evaporation produced a vapor with a quality of 0.1 at the inlet of the flooded evaporator. The refrigerant passed through the flooded evaporator and eventually returned to the condenser. This refrigerant loop contained a condenser, a receiver, a magnetic gear pump, and a pre-heater. The refrigerant flow rate was controlled by a mass flow meter that had a reading accuracy of ±0.1% and was installed between the magnetic gear pump and the pre-heater. The magnetic gear pump was controlled by a variable-speed AC motor, which responded to the frequency changes in the inverter. The pressure of the refrigerant was controlled by adjusting the temperature of the water-glycol mixture in the condenser. Absolute pressure transducers with a measuring range up to approximately 50 kg/cm^2·G and a pressure stability within ±0.075% were installed at the inlet and outlet of the flooded evaporator. A differential pressure transducer was installed on the refrigerant side of the flooded evaporator to measure the overall pressure drop. The temperature of the water and refrigerant was measured by a resistance thermal detection (RTD) sensor (maximum detection limit of 100 °C) with an accuracy of ±0.1%. The overall loop for the water and refrigerant was wrapped with a 3-cm-thick insulated material to reduce the heat loss.

A water-loop-driven water flow at a constant temperature in the range 5-60 °C and with temperature stability within ±0.1% was maintained to feed the flooded evaporator, as shown in Fig. 1. A water pump was used to circulate water at a specific flow rate, and a bypass valve with a calibrated accuracy of ±0.1% was also used. The loop for circulating the mixture of water and glycol was used to supply refrigerant flow at a constant temperature in the range −10-0 °C with temperature stability within ±0.1% to feed the condenser in the plate heat exchanger.

A series of experiments were performed at the refrigerant saturation temperature of 5 °C; moreover, the refrigerant inlet quality was fixed at 0.1 and the heat fluxes were in the range 5-7 kW/m^2, which are inlet conditions for the flooded evaporator. The refriger-

Table 1. Specifications of measuring devices and uncertainty analysis

Parameters	Range	Uncertainty
Temperature	0 to 100 °C	0.1 °C
Pressure	0 to 50 MPa	0.5%
Water flow rate	1,000 to 5,000 kg/h	0.1%
Refrigerant flow rate	90 to 500 kg/h	0.1%
Ambient temperature	0 to 80 °C	0.11 °C
Pressure drop	0 to 50 kPa	0.05 kPa
Heat transfer of refrigerant side	-	2.6%
Heat transfer of water side	-	1.5%

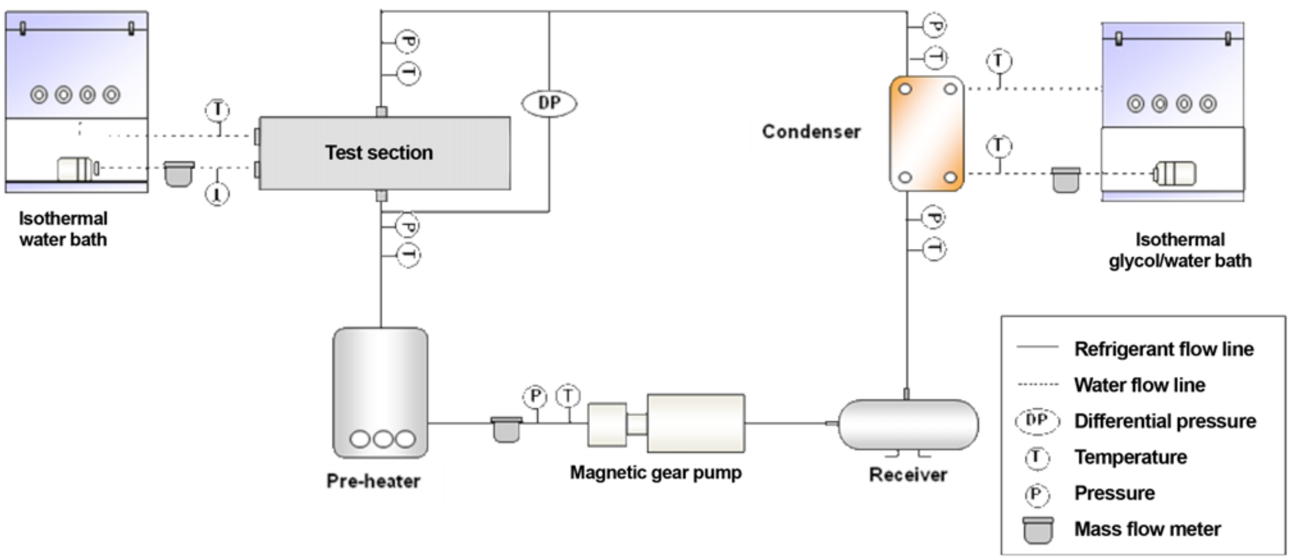

Fig. 1. Schematic of experimental setup.

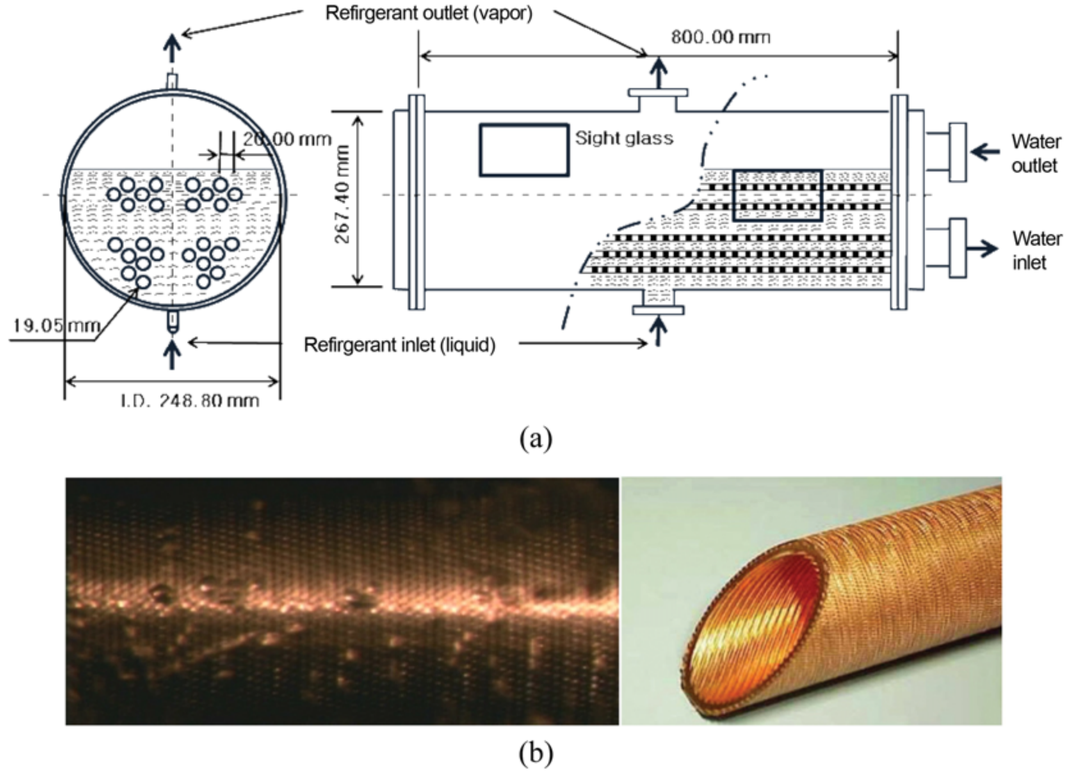

Fig. 2. (a) Schematic of tested shell-and-tube-type flooded evaporator, and (b) cross section of Turbo-E-type tube.

ant saturation temperature of 5 °C was chosen because commercial refrigeration chillers operate at this temperature. The main features of different measurement devices are summarized in Table 1. The uncertainties in the test results were analyzed by the procedures proposed by Kline and McClintock [9].

A schematic diagram of the evaporator used is presented in Fig. 2. In the flooded evaporator, the chilled water flowed inside the tubes and the refrigerant R-134a was boiled on the outside surface of the tubes. The evaporator contained a bundle of enhanced tubes with the following dimensions: length, 800 mm; outer diameter, 19.05 mm; and wall thickness, 1.25 mm. The tube bundle had a triangular pitch configuration and consisted of a total of 26 tubes with a centerline-to-centerline spacing of 37 mm. The number of tube-side passes generally ranges from one to eight. The standard design has one, two, or four tube passes. The flooded evaporator used in this study was manufactured with four passes.

Polyol ester (POE) oil with a viscosity of 175.2 mm^2/s at 40 °C was injected in the flooded evaporator by using an oil pump, which facilitated the control of the oil concentration as desired. Tests were conducted for various oil concentrations of 0, 1, 2, 3, 5, and 10%. The oil concentration in the refrigerant-oil mixture was measured during the experiments to maintain a constant oil concentration. Some samples of refrigerant-oil mixture were taken from the evaporator, and then, the oil concentration was measured by the protocols outlined in ANSI/ASHRAE Standard 41.4-1984. The ANSI/ASHRAE Standard 41.4 is the standard procedure for experimentally determining the weight concentration of miscible lubricant-refrigerant mixtures. Principally, in this method, a refrigerant-oil mixture liquid sample is taken from the system into an evacuated chamber. The liquid sample is then placed in a vacuum environment and heated to 150 °C to evaporate all the refrigerant content. The difference in weight before and after the heating process reflects the amount of refrigerant or oil in the liquid sample.

The boiling heat transfer coefficient (U_O) on the outer tube wall was determined using the following relation:

$$U_O = \frac{q}{A_O \Delta T_{lm}} \tag{1}$$

where ΔT_{lm} is the log mean temperature difference of the evaporator. The heat transfer rate (q) of the shell was determined using the following relation:

$$q = \dot{m}\Delta i \tag{2}$$

where Δi is the decrease in the enthalpy of the tube-side heating fluid across the evaporator and is determined by the decrease in the temperature from the inlet to the outlet.

RESULTS AND DISCUSSION

Fig. 3 shows the variation of the relative heat transfer coefficient of the pure refrigerant and the refrigerant-oil mixture as a function of the oil concentration. It can be observed that the heat transfer was 9% and 11% at oil concentrations of 1% and 10%, respectively; these values were slightly less than that in the case of the pure refrigerant (i.e., for $U_{oil+ref}/U_{ref}=1.0$, see Fig. 3). In other words, the heat transfer degradation of the refrigerant-oil mixture in the flooded evaporator with multiple-inner-tubes-based flooded evaporator was observed to be much lower than that of the refrigerant-oil mixture in a flooded evaporator with a single-inner-tube operated by Kim et al. [5]. The heat transfer degradation in the case of the flooded

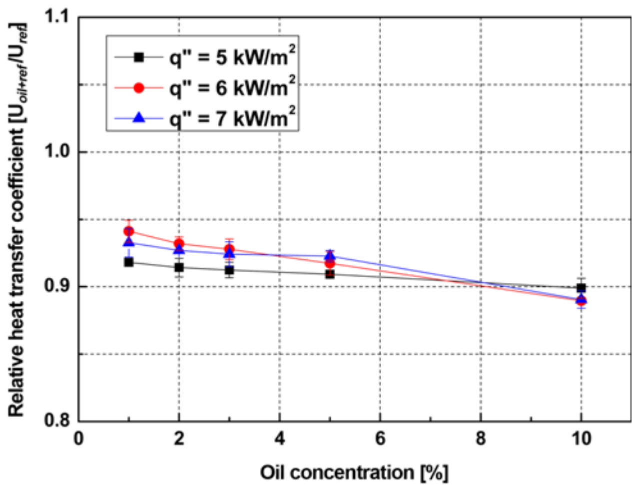

Fig. 3. Effect of oil concentration on relative heat transfer coefficient at various heat fluxes.

evaporator with the single-inner-tube was 26-49%. In general, the heat transfer coefficient of an evaporator is reduced by the presence of lubricant oil in the refrigerant. This is because the lubricant oil can stay on the surface of the inner tube in the flooded evaporator, and results in the formation of a heat-resistant layer; thus, the heat transfer performance of the evaporator is eventually degraded.

In this study, we posed the following question: Why is the heat transfer degradation of a flooded evaporator with multiple-inner-tubes much lower than that of a flooded evaporator with a single-inner-tube? To find an answer, we observed the condition of the refrigerant with various oil concentrations through the windows installed in the evaporator. The results are shown in Fig. 4, which shows the boiling phenomena of the refrigerant-oil mixture in the flooded

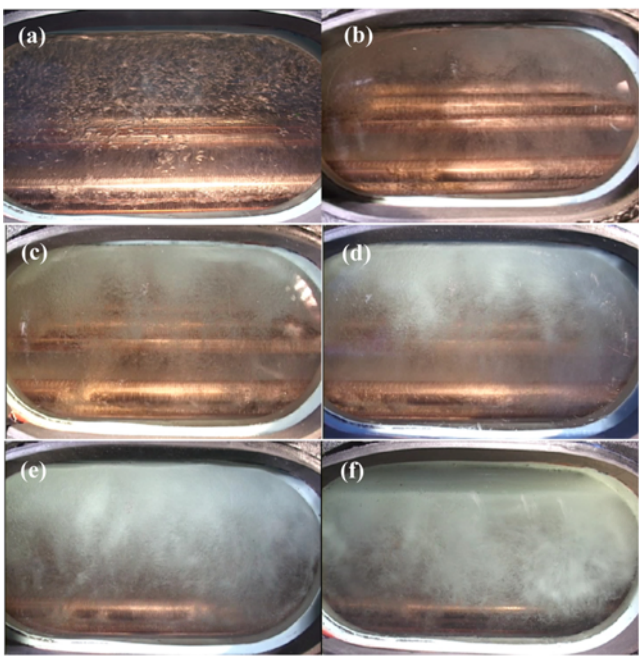

Fig. 4. Variations in boiling of R-134a as a function of oil concentrations of (a) 0, (b) 1, (c) 2, (d) 3, (e) 5, and (f) 10%.

evaporator with multiple-inner-tubes at the refrigerant saturation temperature of 5 °C. Generally, bubbles can form within pre-existing gas pockets located in cracks and imperfections of tube surfaces. It is known as a heterogeneous nucleation process. Briefly, supersaturated gas molecules dissolved in the refrigerant-oil mix liquid diffuse into the gas pockets, eventually causing bubble growth and detachment from the surface of solid tube support. The number of bubble nucleation sites also can be significantly increased with increasing the rough hydrophobic surfaces formed on the tubes. Furthermore, as the oil concentration in the refrigerant-oil mixture was increased in this approach, the total number of oil bubbles (i.e., oil foaming) was also increased significantly. This is because the surface tension of the pure refrigerant increased when the oil concentration increased. This led to an increase in the adhesion between the refrigerant/oil and the surface of the tubes because of which the number of nucleation sites of the bubbles increased. The oil foam floating on top of the liquid refrigerant can degrade the heat transfer performance of the evaporator because the presence of a layer of oil foam near the outlet of the evaporator interferes with the evaporation of the refrigerant.

Unlike the layer of oil foam, the agitating bubbles formed in the refrigerant-oil mixture can promote forced convection heat transfer between inner tubes [10]. This can be corroborated by Chen's correlation [10], which was employed to predict the heat transfer enhancement in a tube bundle:

$$h=Eh_L+h_{mic} \tag{3}$$

where represents the heat transfer coefficient associated with the bulk movement of vapor and liquid. The enhancement factor (E) accounts for the enhancement of the single-phase liquid convective heat transfer coefficient (h_L) because of the turbulence created by the rising vapor bubbles and the impinging of the bubbles on the tube surface. The micro convective component (h_{mic}) accounts for the heat transfer associated with the bubble nucleation and growth. The enhancement factor depends mainly on the agitation due to vapor bubbles rising up from lower tubes and impinging on the tube under consideration. It can be assumed that in the case of a 5×3 tube bundle, E is a function of the boiling number ($B_O=q/h_{fg}\cdot G$), ratio of the pitch to the tube diameter (P/d), row number (N_r=1 for the bottom tube), and column factor (C=1 for single-column tube bundles, C=2 and 3 for the side and central column tubes) [11]. E can be expressed as

$$E=c(B_O)^{m1}(P/d)^{m2}(N_r)^{m3}(C)^{m4} \tag{4}$$

Here, the values of the constant c and the indices m_1, m_2, m_3 and m_4 were determined by using experimental data, and the following expression for the enhancement factor was obtained:

$$E=134.24(B_O)^{0.469}(P/d)^{-0.311}(N_r)^{0.946}(C)^{0.304} \tag{5}$$

A calculation was performed by using Eqs. (3)-(5), and it was observed that the enhancement factors of the side and central tubes were approximately 20% and approximately 32% greater than that of the bottom tube, respectively. Moreover, the enhancement factor was significantly increased upon changing the location from the bottom tube to the top tube; for example, the enhancement factor was observed to have increased by 63% (i.e., when the location was changed from tube bundle row No. 1 to No. 2), 101% (i.e., when the location was changed from tube bundle row No. 1 to No. 3), 128% (i.e., when the location was changed from tube bundle row No. 1 to No. 4), and 149% (i.e., when the location was changed from tube bundle row No. 1 to No. 5). In addition, Chyu et al. [8] experimentally confirmed that the heat transfer coefficient of a refrigerant in tube bundles in an evaporator is increased when the location is changed from the bottom tube to the top tube. This experiment was carried out for a 3×5 tube bundle section. They observed that the heat transfer coefficient of a refrigerant in a tube bundle was much higher than that of a refrigerant in a single tube because of two-phase flow convection. Moreover, the heat transfer coefficient of a refrigerant in a tube bundle in an evaporator increased by up to 20%, 60%, 77%, and 80% when the number of bundle rows was increased from 1 to 2, 3, 4, and 5, respectively. Therefore, it is clear that the increase in the number of tube bundles promoted the bubble agitation, which resulted in an increase in the heat transfer coefficient of the tube bundles in the evaporator.

CONCLUSIONS

We examined the effect of a pure refrigerant and a refrigerant-oil mixture on the heat transfer performance of a flooded evaporator with multiple-inner-tubes, which is actually a shell-and-tube-type heat exchanger. The major results can be summarized as follows: The heat transfer degradation in the case of the flooded evaporator with multiple-inner-tubes was 9% and 11% for oil concentrations of 1% and 10%, respectively. The heat transfer degradation of the flooded evaporator with multiple inner-tubes was half of that of the flooded evaporator with single inner-tube. This observation is primarily attributed to the presence of the bundle of inner tubes immersed in the refrigerant-oil mixture. By liquid visualization, it was found that an increase in the oil concentration in the refrigerant-oil mixture led to increased formation of bubbles and increased oil foaming. It was noted that the agitation associated with the formation of bubbles can significantly enhance the heat transfer, while the oil foaming can reduce the heat transfer by interfering with refrigerant evaporation. The combined effects of bubble formation and oil foaming seemed to eventually decrease the heat transfer degradation in the case of the flooded evaporator with multiple-inner-tubes.

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NOMENCLATURE

A : heat transfer area [m^2]
G : gauge pressure
h : heat transfer coefficient [$W/m^2 \cdot K$]
i : enthalpy [kJ/kg]
Δi : change in enthalpy between the inlet and outlet of tube [kJ/kg]
$\dot{m}$: mass flow rate [kg/h]
q : heat transfer rate [kW]
q'' : heat flux [kW/m^2]
ΔT_{lm} : log mean temperature difference [°C]
U : overall heat transfer coefficient [$kW/m^2 \cdot °C$]

Subscripts

i : inlet
o : outlet
sat : saturation

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