# Effect of Fin Structure on the Condensation of R-134a, R-1234ze(E), and R-1233zd(E) Outside the Titanium Tubes

**Wen-Tao Ji**[1]
Key Laboratory of Thermo-Fluid Science
and Engineering of MOE,
School of Energy and Power Engineering,
Xi'an Jiaotong University,
Xi'an 710049, China
e-mail: wentaoji@xjtu.edu.cn

**Shuai-Feng Mao**
Key Laboratory of Thermo-Fluid Science
and Engineering of MOE,
School of Energy and Power Engineering,
Xi'an Jiaotong University,
Xi'an 710049, China

**Guo-Hun Chong**
Key Laboratory of Thermo-Fluid Science
and Engineering of MOE,
School of Energy and Power Engineering,
Xi'an Jiaotong University,
Xi'an 710049, China

**Chuang-Yao Zhao**
Key Laboratory of Thermo-Fluid Science
and Engineering of MOE,
School of Energy and Power Engineering,
Xi'an Jiaotong University,
Xi'an 710049, China

**Hu Zhang**
Key Laboratory of Thermo-Fluid Science
and Engineering of MOE,
School of Energy and Power Engineering,
Xi'an Jiaotong University,
Xi'an 710049, China

**Wen-Quan Tao**
Key Laboratory of Thermo-Fluid Science
and Engineering of MOE,
School of Energy and Power Engineering,
Xi'an Jiaotong University,
Xi'an 710049, China

*In order to test the effect of fin structure on the condensing heat transfer of refrigerants outside the low thermal conductivity tubes, condensation of R-134a, R-1234ze(E), and R-1233zd(E) on two enhanced titanium tubes were experimentally investigated. The two tubes have basically the same fin density while the fin structures are different. One tube is a typical low-fin (two-dimensional, 2D), and the other is a three-dimensional (3D) finned tube. In experiment heat flux was in the range of 10–80 kW·m$^{-2}$. It was found that at higher heat flux, the condensing heat transfer coefficient (HTC) of 3D-finned tubes was apparently lower than that of 2D-enhanced tubes. The condensing HTC of R-134a for the two tubes was the highest. R-1233zd(E) was the lowest. It was shown from experimental results that the condensing HTC for R-1233zd(E) was notably affected by the change of saturation temperature outside the 3D-enhanced tube, but was less affected by the 2D fin structures.* [DOI: 10.1115/1.4045139]

*Keywords: condensation, new refrigerants, titanium tube*

## 1 Introduction

Shell and tube condensers are widely used in heating, cooling, and a variety of industrial applications. The requirement on more compact and effectiveness results in the increased demand for higher condensing heat transfer coefficient (HTC). Many studies focused on optimizing the structures of condensing tubes [1,2]. Effect of materials [2–4] was also investigated. Some of them [5] also committed to studying new refrigerants with the characteristics of environment-friendly, safe, and superior thermophysical properties.

Titanium is a good material that can be used under some extreme conditions. Nowadays, the main problem pertaining to the use of titanium tubes is the poor condensing heat transfer performance. Further improving the condensing heat transfer of titanium tubes will be worthy of study. Whether the three-dimensional structure has great effect on the condensing heat transfer as that for copper tubes, condensation of refrigerants on two different enhanced titanium tubes were experimentally investigated. The two tubes have the same fin density while the fin structures are completely different. One tube is a typical low-fin (two-dimensional, 2D) and the other is a three-dimensional (3D) finned tube. The effect of fin structure for the titanium tubes was also tested with the environment-friendly refrigerants R-1234ze(E) and R-1233zd(E).The experimental system, procedure, and data reduction are the same as that in Ref. [4]. Figure 1 shows the schematic diagram of experimental apparatus and photos of tubes in condensation at different heat flux.

The specific geometric parameters for two tubes are shown in Table 1. Figure 2 is the geometric structure of two enhanced tubes. The method in Refs. [4] and [6–8] is used to estimate the uncertainty of experiment. The uncertainty for $A_o$ is less than 0.6%; $q$ is less than 5.4%; $k$ is less than 5.5%. The uncertainty for Gnielinski equation is estimated for 10% according to Refs. [9–12]. Then, the measurement uncertainty for external HTC $h_o$ is less than 15.1%.

## 2 Results and Discussion

**2.1 Reliability Verification of Experimental System.** Experiment for plain tube is first conducted to validate the experimental apparatus. The condensing heat transfer coefficient for the plain tube is compared with the Nusselt analytical solution. Nusselt analytical solution [13,14] is shown as follows:

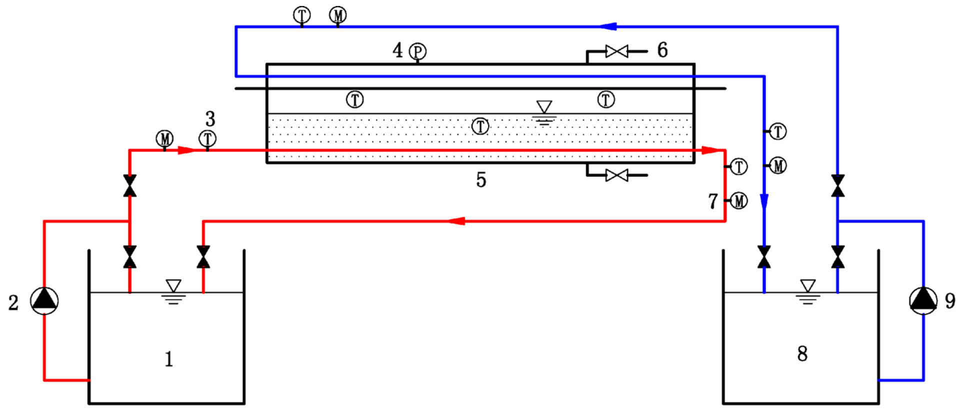
$$h_o = 0.729\left(\frac{rg\lambda_l^3\rho_l^2}{\eta_l d_o(t_s - t_w)}\right)^{\frac{1}{4}} = 0.656\left(\frac{rg\lambda_l^3\rho_l^2}{\eta_l d_o q}\right)^{\frac{1}{3}} \quad (1)$$

As shown in Fig. 3, the relative deviations are less than 7% for experimental results of R-134a and Nusselt analytical solution over the plain titanium tube. From the experimental results on the plain titanium tube, it is demonstrated that the experimental results are reliable for the present experimental system and procedures.

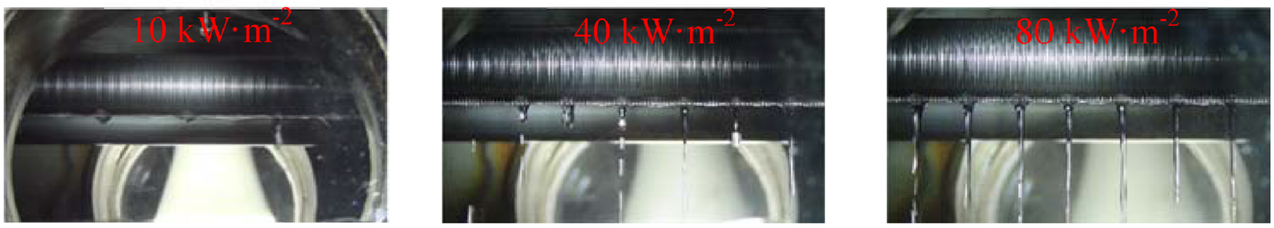
[1]Corresponding author.

Contributed by the Heat Transfer Division of ASME for publication in the Journal of Heat Transfer. Manuscript received June 9, 2019; final manuscript received September 25, 2019; published online November 8, 2019. Assoc. Editor: Guihua Tang.

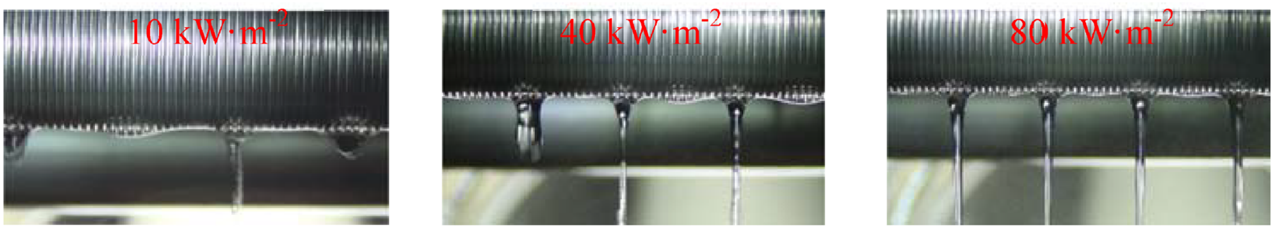
  




(*a*)




(*b*)




(*c*)

**Fig. 1 Schematic diagram of the experimental apparatus and visualization observation: (*a*) schematic diagram of experimental apparatus: (1) hot water tank, (2) hot water pump, (3) platinum resistance thermometer, (4) digital pressure gauge, (5) condenser, (6) exhausting valve, (7) electromagnetic flow meter, (8) cold water tank, (9) cold water pump; (*b*) global view; (*c*) local view**

**Table 1 Specifications of test tubes**

| Tubes | Outside diameter, $d_o$ (mm) | Inside diameter, $d_i$ (mm) | Thickness of outside fin, $\delta$ (mm) | Height of outside fin, $e$ (mm) | Outside fins per inch |
|---|---|---|---|---|---|
| Plain | 15.99 | 14.85 | — | — | — |
| 2D enhanced (tube No.1) | 16.03 | 13.96 | 0.424 | 0.639 | 31 |
| 3D enhanced (tube No.2) | 16.07 | 14.01 | 0.658–0.782 | 0.410–0.458 | 31 |

**2.2 Overall Heat Transfer Coefficient.** Overall HTC for R-134a, R-1234ze(E), and R-1233zd(E) outside the 2D- and 3D-finned titanium tubes are obtained first. Experimental results for the overall HTC of three refrigerants are shown in Figs. 4 and 5.

Overall HTC versus water velocity for the 2D-enhanced titanium tube (No. 1) is shown in Fig. 4. As shown in Figs. 4(*a*)–4(*c*), overall HTC of R-134a is the highest. It is 2.6–11.6% higher than R-1234ze(E), and about 10.2–25.3% higher than R-1233zd(E) under the same conditions for 2D-enhanced titanium tube. Figure 5 is the overall heat transfer coefficient of R-134a, R-1234ze(E), and R-1233zd(E) versus water velocity for 3D finned tube (No. 2). As shown in Figs. 5(*a*)–5(*c*) for 3D-enhanced titanium tube, the overall heat transfer coefficient of R-134a is also the highest. It is approximately 1.3–14.5% higher than R-1234ze(E), and 14.6%-35.4% higher than R-1233zd(E) with the same experimental condition.

For the 3D-enhanced tubes, the overall heat transfer coefficient of R-134a, R-1234ze(E), and R-1233zd(E) all gradually decreased with increasing of heat flux at same water velocity. However, for the 2D-enhanced tube, the effect of heat flux on the overall HTC is not obvious for all the three refrigerants. For different refrigerants and different flux, the overall HTCs of the two tubes are also

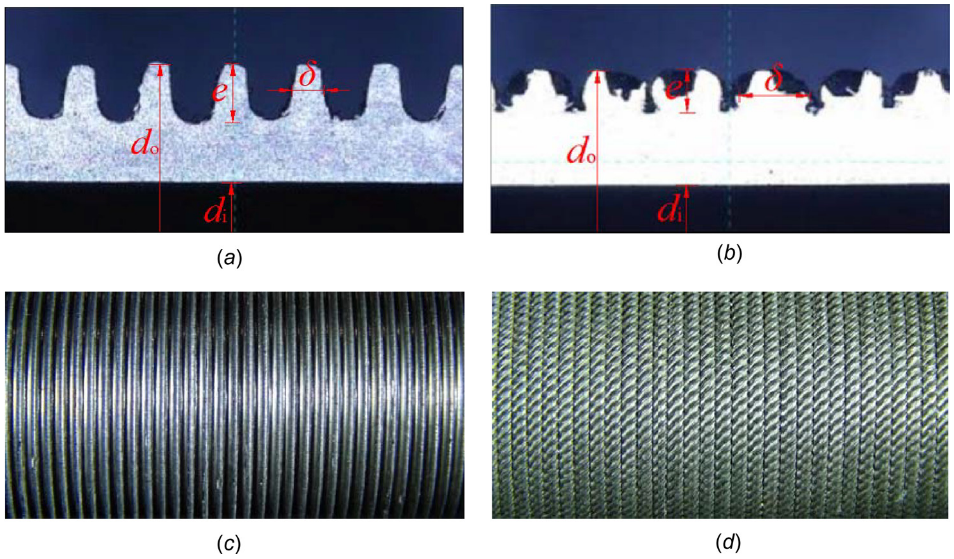



**Fig. 2 Geometric structure of enhanced titanium tubes: (*a*) 2D enhanced tube (No. 1), (*b*) 3D enhanced tube (No. 2), (*c*) 2D tube (global view), and (*d*) 3D tube (global view)**

different. For R-134a, the two tubes with different structures have basically the same performance. For R-1233ze(E) and R-1233zd(E), at the lower heat flux, HTC of 3D-finned tubes is larger than the 2D-enhanced tubes. As the increase of heat flux, the overall HTC of 3D-enhanced tubes are decreasing. At the heat flux of 40 kW/m$^2$, for R-1234ze(E) and R-1233zd(E), the 2D-finned tubes both have higher overall HTC than the 3D-finned tubes.

**2.3 Condensing Heat Transfer Coefficient.** Condensing HTC versus heat flux is shown in Fig. 6. The results are described and discussed as follows:

(1) For all the three refrigerants, 3D-enhanced tube has higher condensing heat transfer coefficient at lower heat flux and lower heat transfer coefficient at higher heat flux. The turning point is at 20 kW/m$^2$. As the increase of heat flux, the decreasing rate of 3D-enhanced tube is higher than the 2D-finned tube. It indicates that at lower heat flux, due to the thinning of condensate film by the effect of surface tension, the three-dimensional structure is beneficial to the condensing heat transfer of titanium tubes. While as the increase of heat flux, the condensate outside the fins is also increasing. For the three-dimensional fins, the fin structure is complicated and rather irregular. It tends to retain more condensate outside the finned structures. The amount of liquid that being retained between the fins should be more than the 2D-enhanced tubes. The flooded area should also be different. Condensate retention will reduce the condensing heat transfer of finned tubes. This is the reason that 2D-enhanced tubes has higher heat transfer coefficient than 3D-finned tubes at higher heat flux.

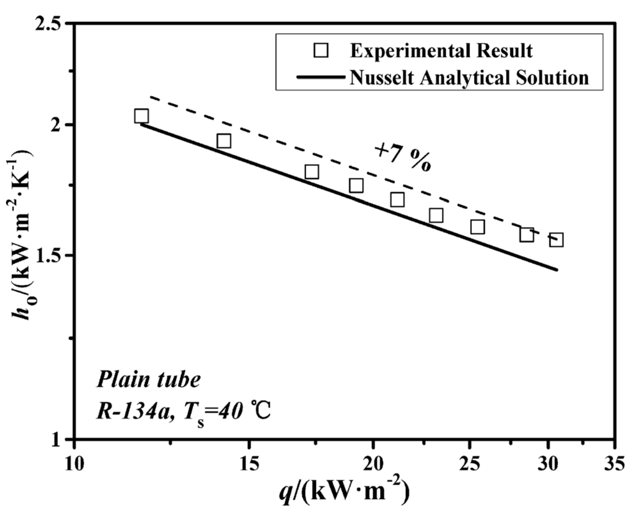



**Fig. 3 Comparison between experimental result and Nusselt analytical solution**

(2) The enhancement of heat transfer for the 3D-finned structures over 2D-finned titanium tubes is limited. Compared with copper tubes, the enhancement of 3D fins over the 2D-enhanced fins for titanium tubes with the same fin density is not obvious. For the copper finned tubes with the same fin density, the condensing heat transfer coefficient is obviously higher than that made by the material with lower thermal conductivity [4]. In this study, the condensing heat transfer coefficient of 3D-enhanced tubes is only 1–9% higher than the 2D-enhanced tubes at lower heat flux. At the higher heat flux, the heat transfer coefficient is even less than the 2D low-fin tubes. For the copper tubes in Refs. [15] and [16], this enhancement is in the range of 20–30% within the heat flux of 10–80 kW/m$^2$. For copper tubes, the decreasing rate for the condensing heat transfer coefficient as the increase of heat flux is not as sharp as the titanium tubes [15,16].

(3) For the refrigerant R-1233zd(E), as the increase of the heat flux, the condensing heat transfer coefficient of 2D- and 3D-enhanced tubes first increase and then decrease. The critical heat flux is also at 20 kW/m$^2$. It can be seen in Figs. 2(*a*) and 2(*b*) that the circumferential groove is deeper between two fins for tube No. 1 comparing to tube No. 2. As heat flux is below 20 kW·m$^{-2}$, flow rate of liquid film is limited. Liquid refrigerant might be kept in the deep groove, which will result in the growing of liquid film thickness and being detrimental to the condensing heat transfer performance.

(4) The condensing HTC gradually decreases with increasing of heat flux for R-134a and R-1234ze(E) on 2D-enhanced tube No. 1. Comparing the condensing HTC for testing refrigerants on tube No. 1, condensing HTC of R-134a is the highest at the same

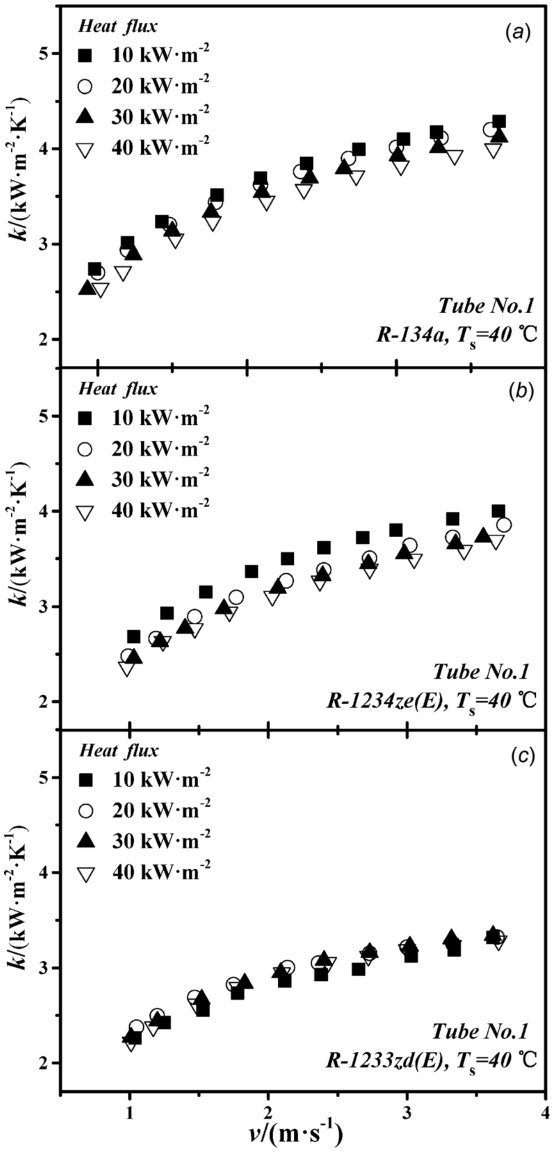



**Fig. 4 Overall HTC versus water velocity at different heat flux for tube No.1 (2D enhanced tube)**

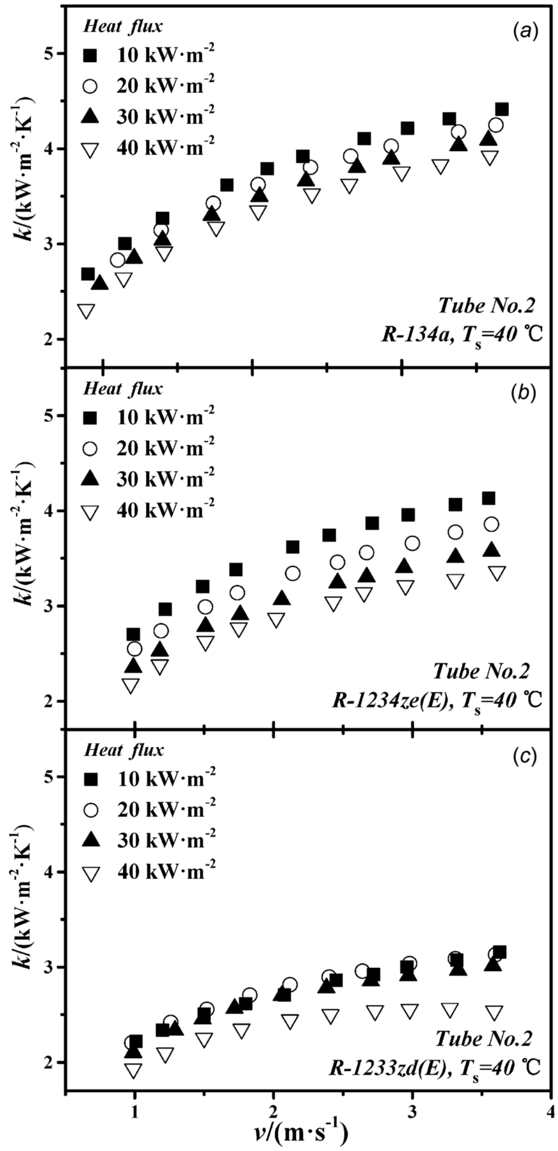



**Fig. 5 Overall HTC versus water velocity at different heat flux for tube No.2 (3D enhanced tube)**

experiment condition. Figures 6(*c*) and 6(*d*) show the condensing HTC versus heat flux for tube No. 2. Condensing HTC is also highest for R-134a, and R-1233zd(E) is lowest for tube No. 2. As shown in the figure, the optimum fin structure for the heat transfer performance is dependent upon the refrigerant. Compared with the investigation in the literature, condensing heat transfer for R-1233zd(E) are all less than R-134a and R-1234ze(E) [17]. The heat transfer coefficient of R-1234ze(E) is a little bit lower than R-134a. It is consistent for the present investigation and that from literature.

As shown in Figs. 2(*a*) and 2(*b*) and Table 1, fin height for tube No. 2 is lower than tube No. 1, and fin density for tube No. 2 is identical with tube No. 1 in the axial direction. Because the fins for No. 2 is three-dimensional, three-dimensional mixing and mass transfer in liquids is different with No. 1. The three-dimensional enhanced tube will diminish the heat transfer performance when the film condensate is thick. The Gregorig effect for titanium tubes is not as effective as the copper tubes. It is generally consistent with the result in the literature [4].

### 2.4 Effect of Saturation Temperature on Condensing Heat Transfer Coefficient.

Figure 7 shows the effect of saturation temperature on condensing HTC of R-134a, R-1234ze(E), and R-1233zd(E) over the 2D and 3D enhanced tubes. Saturation temperature is ranging from 35 to 40 °C.

For the 2D-enhanced tube, the influence of saturation temperature on condensing HTC can be almost negligible for three testing refrigerants. Comparing with the 2D-enhanced tube, Fig. 7(*b*) shows that condensing HTC is less affected by the change of saturation temperature for R-134a and R-1234ze(E) outside the 3D-enhanced tube. However, condensing HTC of R-1233zd(E) is apparently affected by saturation temperature for the 3D-enhanced tube. Condensing HTC decreases obviously for R-1233zd(E) as saturation temperature decreased. As mentioned previously, due to the difference in manufacturing process, outer surface area of 3D enhanced tube No.2 is larger than tube No. 1. Liquid viscosity increased with decreasing saturation temperature for R-1233zd(E). Then thickness of liquid film for saturation temperature of 35 °C should be thicker than 40 °C, a corresponding decrease in condensing HTC should also be observed for R-1233zd(E) on 3D-enhanced tube No.2 at saturation temperature of 35 °C.

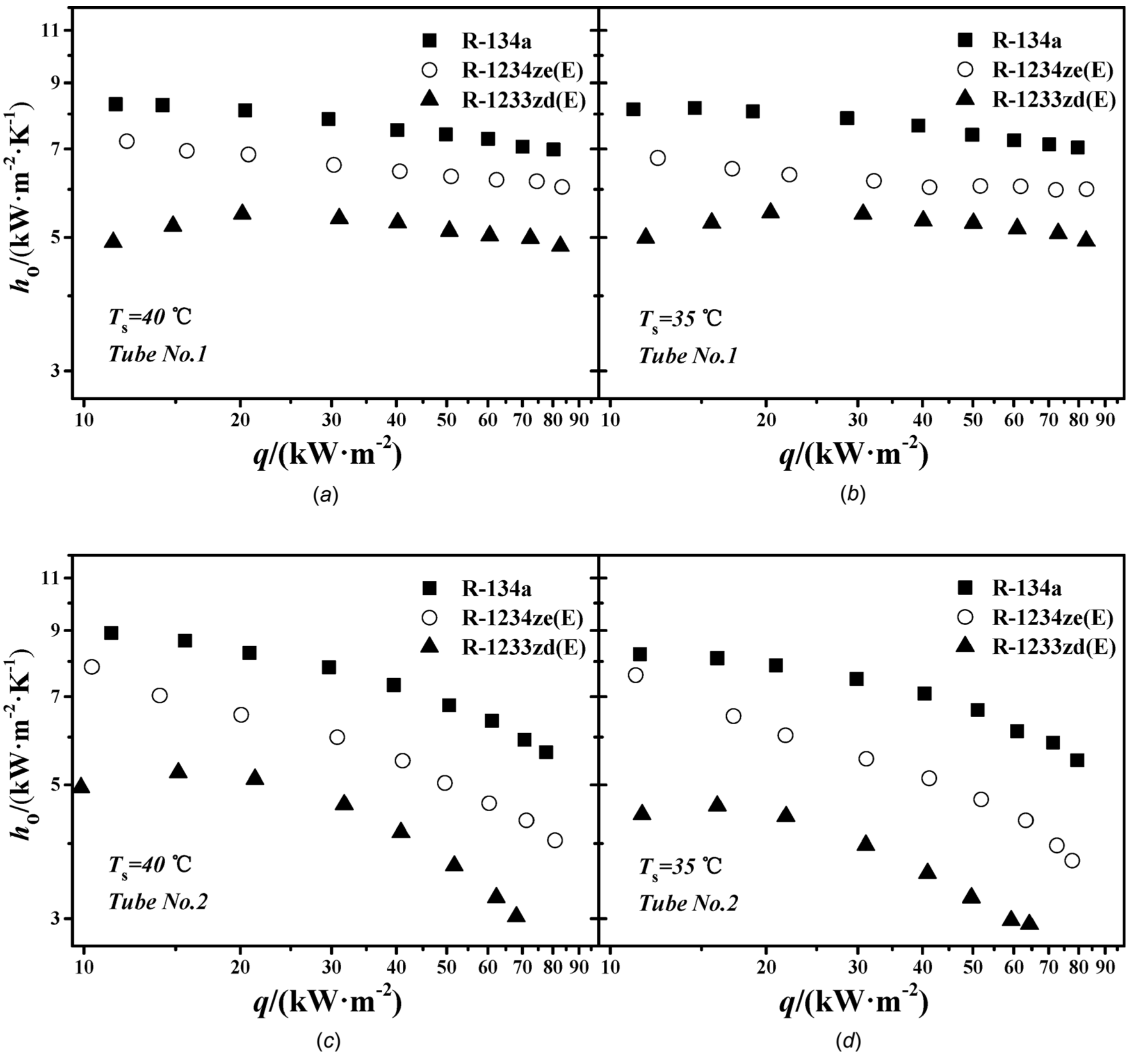



**Fig. 6 Condensing HTC versus heat flux at saturation temperature of 40 °C and 35 °C on tube nos. 1 and 2**

As shown above, condensing heat transfer of two different types of titanium tubes are presented. It is helpful for the designers to summarize the condensing heat transfer performance of some new refrigerants. The 3D-enhanced tubes generally have the similar condensing heat transfer coefficients as the low-fin tubes with same external fin density at lower heat flux. Generally, the decreasing rate of condensing HTC for the 3D-enhanced tube with increasing heat flux is higher. Because of higher viscosity, the heat transfer performance for R-1233zd(E) is decreasing for all testing enhanced tubes at lower heat flux less than 20 kW·m$^{-2}$. However, the trend of variations of the performance on R-134a and R-1234ze(E) is similar.

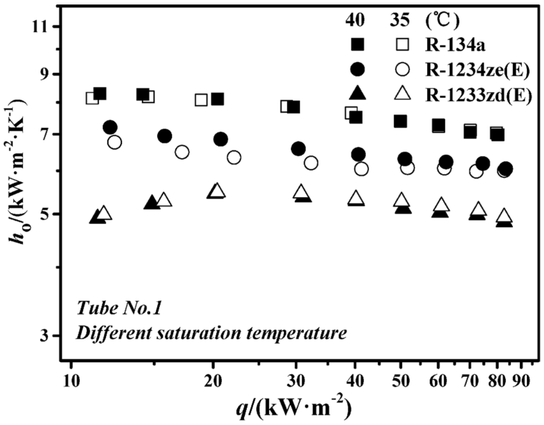
There are plenty of data available for copper tube with similar geometries and operating conditions [3,4]. However, the result seems to be not conforming with the tubes of lower thermal conductivity. That is the reason that this study with different types of fin structure is conducted. The two testing tubes have the same external fin density but different fin structure. Fin height of 2D-enhanced tubes is higher than 3D-enhanced tube. Normally, larger fin height might contribute to higher HTC over the higher thermal conductivity tube [18]. While according to the investigation, the result shows that it is different for different tube and refrigerant. For R-134a and R-1234ze(E), the 3D-enhanced tube has a higher HTC. While, for R-1233zd(E), 2D-enhanced tube has higher HTC than 3D enhanced tube. The reason might be the effect of thermal properties of refrigerants. The surface tension for R-1233zd(E) is about 2 times higher than those of other two refrigerants. The viscosity is also almost 50% higher. It can be inferred that the improvement in condensing HTC depends both on refrigerant and fin structures.

## 3 Conclusions

Condensation of R-134a, R-1234ze(E), and R-1233zd(E) outside 2D (No.1) and 3D (No.2) enhanced titanium tubes were experimentally investigated. Saturation temperature was in range of 35–40 °C, heat flux was within 10–80 kW·m$^{-2}$. Based on the experimental result, the following conclusion can be drawn from this study:

(1) Overall heat transfer coefficient for R-134a is the highest, and R-1233zd(E) is the lowest for the 2D- and 3D-finned tubes. As heat flux decreased to 10 kW·m$^{-2}$, overall heat transfer coefficient of R-1233zd(E) sharply decreases for tube No.1. The thermal resistance of tube wall for titanium tube is approximately four times higher than that of copper tube.

(2) The 3D-enhanced tube generally has higher condensing heat transfer coefficient at lower heat flux and lower heat transfer coefficient at higher heat flux. The critical heat flux is at 20 kW/m$^2$. Condensing heat transfer coefficient for R-134a is the highest, and R-1233zd(E) is the lowest for the tubes with different types of structures.



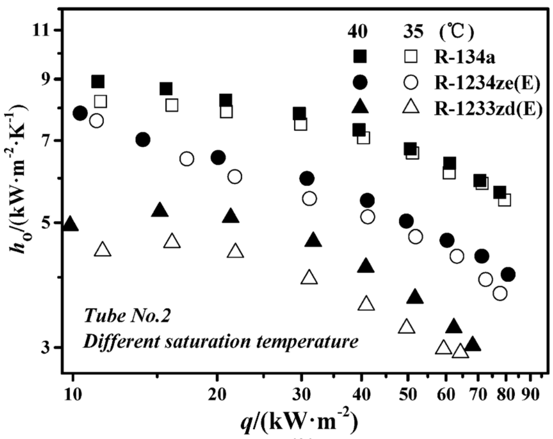




**Fig. 7 Effect of saturation temperature on condensing HTC**

(3) Compared with copper tubes, the enhancement ratio of 3D fins over 2D fins with the same fin density is not obvious. Gregorig effect for titanium tubes is not as effective as the copper tubes.

(4) Condensing heat transfer coefficient is less affected by the change of saturation temperature for R-134a, R-1234ze(E) on the 2D- and 3D-finned tubes. However, the effect is notable for R-1233zd(E) outside the 3D-enhanced tube.

## Funding Data

- National Natural Science Foundation of China (51776160; Funder ID: 10.13039/501100001809).

## Nomenclature

$A$ = area (m$^2$)
$c_p$ = specific heat capacity (J·kg$^{-1}$·K$^{-1}$)
$d$ = diameter of tube (mm)
$e$ = height of outside fin (mm)
$f$ = drag coefficient
$g$ = gravitational acceleration (m·s$^{-2}$)
$h$ = heat transfer coefficients (W·m$^{-2}$·K$^{-1}$)
$k$ = overall heat transfer coefficients (W·m$^{-2}$·K$^{-1}$)
$L$ = tube's test length (m)
$m$ = mass flow rate (kg·s$^{-1}$)
Pr = Prandtl number in Gnielinski equation
$q$ = heat flux (W·m$^{-2}$)
$R_f$ = thermal resistance of foul (m$^2$·K·W$^{-1}$)
$R_w$ = thermal resistance of tube wall (m$^2$·K·W$^{-1}$)
Re = Reynolds number
$t$ = temperature (°C)

### Greek Symbols

$r$ = latent heat (kJ·kg$^{-1}$)
$\Delta t_m$ = logarithmic mean temperature difference (K)
$\eta$ = viscosity (Pa·s)
$\lambda$ = thermal conductivity (W·m$^{-1}$·K$^{-1}$)
$\rho$ = density (kg·m$^{-3}$)
$\sigma$ = surface tension (mN·m)
$\phi$ = heat transfer rate (W)

### Subscripts

b = boiling
c = condensing
i = inside of tube
in = inlet of tube
ip = inside of plain tube
l = liquid
o = outside of tube
out = outlet of tube
p = plain
s = saturation
w = wall

## References

[1] Al-Badri, A. R., Bär, A., Gotterbarm, A., Rausch, M. H., and Fröba, A. P., 2016, "The Influence of Fin Structure and Fin Density on the Condensation Heat Transfer of R134a on Single Finned Tubes and in Tube Bundles," Int. J. Heat Mass Transfer, **100**, pp. 582–589.
[2] Zhao, C.-Y., Ji, W.-T., Jin, P.-H., Zhong, Y.-J., and Tao, W.-Q., 2017, "The Influence of Surface Structure and Thermal Conductivity of the Tube on the Condensation Heat Transfer of R134a and R404A Over Single Horizontal Enhanced Tubes," Appl. Therm. Eng., **125**, pp. 1114–1122.
[3] Zhang, D. C., Ji, W. T., and Tao, W. Q., 2007, "Condensation Heat Transfer of HFC134a on Horizontal Low Thermal Conductivity Tubes," Int. C. Heat Mass Transfer, **34**(8), pp. 917–923.
[4] Ji, W.-T., Zhao, C.-Y., Zhang, D.-C., Li, Z.-Y., He, Y.-L., and Tao, W.-Q., 2014, "Condensation of R134a Outside Single Horizontal Titanium, Cupronickel (B10 and B30), Stainless Steel and Copper Tubes," Int. J. Heat Mass Transfer, **77**, pp. 194–201.
[5] Jung, D., Kim, C.-B., Hwang, S.-M., and Kim, K.-K., 2003, "Condensation Heat Transfer Coefficients of R22, R407C, and R410A on a Horizontal Plain, Low Fin, and Turbo-C Tubes," Int. J. Refrig., **26**(4), pp. 485–491.
[6] Kline, S. J., 1953, "The Description of Uncertainties in Single Sample Experiments," Mech. Eng., **75**, pp. 3–9.
[7] Ji, W.-T., Numata, M., He, Y.-L., and Tao, W.-Q., 2015, "Nucleate Pool Boiling and Filmwise Condensation Heat Transfer of R134a on the Same Horizontal Tubes," Int. J. Heat Mass Transfer, **86**, pp. 744–754.
[8] Lienhard, J. H., 2013, *A Heat Transfer Textbook*, Courier Corporation, Cambridge, Massachusetts, U.S.A.
[9] Gnielinski, V., 1976, "New Equations for Heat and Mass Transfer in Turbulent Pipe and Channel Flow," Int. Chem. Eng., **16**(2), pp. 359–368.
[10] Yunus, A. C., and Ghajar, A. J., 2007, *Transferencia de Calor y Masa*, Segunda Edición, Editorial Mc GrawHill, México.
[11] Bergman, T. L., Incropera, F. P., Lavine, A. S., and Dewitt, D. P., 2011, *Introduction to Heat Transfer*, Wiley, Danvers, MA, U.S.A.
[12] Ji, W.-T., Zhao, C.-Y., Lofton, J., Li, Z.-Y., Zhang, D.-C., He, Y.-L., and Tao, W.-Q., 2018, "Condensation of R134a and R22 in Shell and Tube Condensers Mounted With High-Density Low-Fin Tubes," ASME J. Heat Transfer, **140**(9), p. 091503.
[13] Nusselt, W., 1916, "Dia Oberflachen Condensation Des Wasserdampfes," Zeitchrift Des Vereines Deustcher Ingenieure, **60**, pp. 1645–1648.
[14] Dhir, V., and Lienhard, J., 1971, "Laminar Film Condensation on Plane and Axisymmetric Bodies in Nonuniform Gravity," ASME J. Heat Transfer, **93**(1), pp. 97–100.
[15] Honda, H., Uchima, B., Nozu, S., Torigoe, E., and Imai, S., 1992, "Film Condensation of R-113 on Staggered Bundles of Horizontal Finned Tubes," ASME J. Heat Transfer, **114**(2), pp. 442–449.
[16] Kumar, R., Varma, H. K., Mohanty, B., and Agrawal, K. N., 2002, "Augmentation of Heat Transfer During Filmwise Condensation of Steam and R-134a Over Single Horizontal Finned Tubes," Int. J. Heat Mass Transfer, **45**(1), pp. 201–211.
[17] Nagata, R., Kondou, C., and Koyama, S., 2016, "Comparative Assessment of Condensation and Pool Boiling Heat Transfer on Horizontal Plain Single Tubes for R1234ze (E), R1234ze (Z), and R1233zd (E)," Int. J. Refri., **63**, pp. 157–170.
[18] Ji, W.-T., Zhao, C.-Y., Zhang, D.-C., He, Y.-L., and Tao, W.-Q., 2012, "Influence of Condensate Inundation on Heat Transfer of R134a Condensing on Three Dimensional Enhanced Tubes and Integral-Fin Tubes With High Fin Density," Appl. Therm. Eng., **38**, pp. 151–159.