CHEMICAL ENGINEERING TRANSACTIONS

VOL. 88, 2021

A publication of AIDIC
The Italian Association of Chemical Engineering
Online at www.cetjournal.it

Guest Editors: Petar S. Varbanov, Yee Van Fan, Jiří J. Klemeš

**ISBN** 978-88-95608-86-0; **ISSN** 2283-9216

DOI: 10.3303/CET2188114

# Numerical Study of Convective Condensation Heat Transfers for Moist Air on a 3-D Finned Tube

Yuheng Gu, Yudong Ding*, Qiang Liao, Qian Fu, Xun Zhu, Hong Wang

Chongqing University, Key Laboratory of Low-grade Energy Utilization Technologies and Systems, Shazheng St.174, Shapingba Dist., Chongqing 400030, China
Corresponding author: dingyudong@cqu.edu.cn

The condensation of moist air is a common process in industry and increase its heat transfer coefficient is of great importance for improving energy efficiency. This study used 3-D finned tube to enhance moist air condensation, and the condensation process was calculated through numerical method aligned with Diffusion Layer model and Eulerian Wall Film model. The total heat transfer, sensible heat transfer and latent heat transfer during the condensation process were discussed. The results indicated that 3-D finned tube increased heat transfer amount and the condensation was mainly on fin surface. Besides, the influence of 3-D fin height was also obtained. The results indicated that the increase of fin height in circumferential direction can increase heat transfer in the tail of tube, and sensible heat transfer process was more sensitive to the area increment.

## 1. Introduction

The condensation of moist air is a common process in the heat and mass transfer background, such as gas fired boiler, seawater desalination, $CO_2$ capture and storage and so on. For example, in gas fired boiler, the exhaust gas temperature is about 150 °C and the dew point is about 50 °C. The energy coefficient of boiler can be improved greatly if latent heat can be recovered. 3-D finned tube is an effective heat transfer enhancement element and it has been applied in industry for a long time. However, the enhancement of 3-D finned tube for moist air has not been studied.

During the condensation processes, noncondensable gases accumulate near the tube. This decreased the partial pressure of steam and decreased the driving force of condensation, making it difficult for steam to condense on cooling wall. The condensation characteristics of steam in the presence of noncondensable gases has been studied by many researchers. For example, Othmer (1929) studied the condensation of steam in the presence of noncondensable gas and found that 0.5 % mass fraction of NCG can decrease the heat transfer more than 50 %. Sparrow and Lin (1964) developed boundary layer models to the study the influence of NCG and found satisfactory agreement with experimental result. Tang et al. (2012) developed a double boundary layer to model the film condensation of steam in the presence of NCG.

Adding fins to heat transfer surface is an effective method to enhance heat transfer. Fins can increase heat transfer area and break flow boundary near tube wall. Chen et al. (2018) studied the heat transfer and pressure drop outside heat exchanger and found 34 % improvements of *Eu* number. Gu et al. (2020) studied moist air condensation outside 3-D finned tubes with different wettability. They found that hydrophilic finned tube has the best enhancement result. Tong et al. (2015) studied the condensation of moist air outside a pin fin tube under gravity driven flow and found that fin pin tube can enhance heat transfer under higher NCG condition. Besides, Ge (2018) studied the condensation of steam with a large amount of $CO_2$ outside a V-Shaped Plate and found that a proper increase in fin spacing can enhance the condensation process and gold plating has small influence on the condensation process.

To improve the heat transfer performance of 3-D finned tube. Numerical method was applied in this study. The influence of 3-D fin height was also obtained. The latent and sensible heat flux and heat transfer coefficient were obtained and analysed. The results obtained in this study can give reference for the design of heat transfer enhancement used in moist air condensation.

Paper Received: 21 June 2021; Revised: 24 August 2021; Accepted: 14 October 2021
Please cite this article as: Gu Y., Ding Y., Liao Q., Fu Q., Zhu X., Wang H., 2021, Numerical Study of Convective Condensation Heat Transfers for Moist Air on a 3-D Finned Tube, Chemical Engineering Transactions, 88, 685-690 DOI:10.3303/CET2188114

## 2. Physical model and numerical method

In this section, the physical model studied was introduced at first. Models used to calculate the condensation process was presented and boundary settings were also listed. After that, mesh validation was carried out and the result was compared with previous studies. Then, 3-D finned tubes with different fin height was studied and results were obtained and analyzed.

### 2.1 Physical model

The model of 3-D finned tube was shown in Figure 1. Due to the symmetry of the tube, only half the tube is shown in the figure. 3-D fins was numbered along circumferential direction, from F1 to F8. The width of fin is 2.0 mm and the circumferential fin spacing is 2.5 mm. The outside and internal tube diameter were 20.0 mm and 17.0 mm, respectively. The length in axial direction is 1.5 mm. The height of 3-D fins was listed in Table 1.

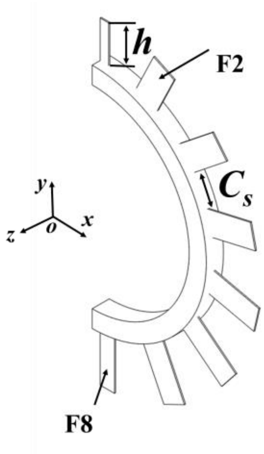



*Figure 1: Schematic view of 3-D finned tube*

*Table 1: The height of 3-D fins (mm)*

| 3-D fin number | Tube No. 1 | Tube No. 2 | Tube No. 3 |
|---|---|---|---|
| F1 | 3.50 | 3.50 | 3.50 |
| F2 | 3.68 | 3.85 | 4.20 |
| F3 | 3.86 | 4.24 | 5.04 |
| F4 | 4.05 | 4.66 | 6.05 |
| F5 | 4.25 | 5.12 | 7.26 |
| F6 | 4.47 | 5.64 | 8.71 |
| F7 | 4.69 | 6.20 | 10.45 |
| F8 | 4.92 | 6.82 | 12.54 |

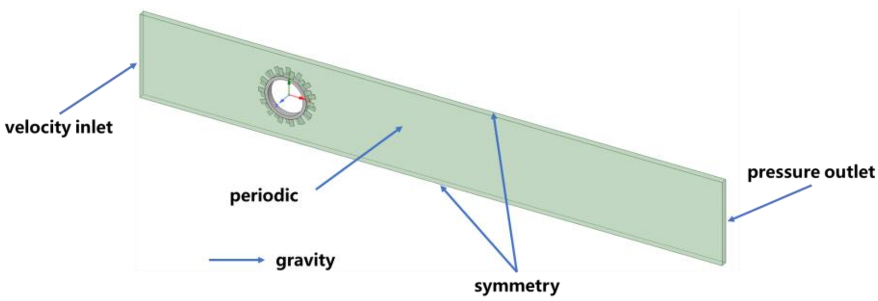



*Figure 2: Schematic view of boundary setup*

The boundary settings were shown in Figure 2. The inlet and outlet of the drainage basin were respectively set as velocity inlet and pressure outlet, where the inlet temperature was 333 K, the velocity was 2.4 m/s, and the volume fraction of water vapor was 0.18. The inner wall temperature of the tube was set as a constant wall temperature boundary with a temperature of 318.8 K. The upper and lower boundary is set as symmetric boundary, and the axial side of the heat exchange tube is set as periodic boundary condition.

### 2.2 Governing equations and boundary conditions

The governing equations were listed from Eq(1) to Eq(13). The mass conservation equation:

$$\frac{\partial \rho}{\partial t}+\nabla\cdot(\rho\vec{v})=S_m \tag{1}$$

where $S_m$ is source term. The momentum conservation equation:

$$\frac{\partial}{\partial t}(\rho\vec{v})+\nabla\cdot(\rho\vec{v}\vec{v})=-\nabla p+\nabla\cdot(\bar{\bar{\tau}})+\rho g+\vec{F} \tag{2}$$

where $\tau$ is shear stress, and is calculated as follow:

$$\bar{\bar{\tau}}=\mu[(\nabla\cdot\vec{v}+\nabla\vec{v}^{T})-\frac{2}{3}\nabla\cdot\vec{v}I] \tag{3}$$

Energy conservation equation:

$$\frac{\partial(\rho T)}{\partial t}+\frac{\partial(\rho u_i T)}{\partial x_i}=\frac{\partial}{\partial x_i}[\frac{\lambda}{C_p}\frac{\partial T}{\partial x_i}]+S_T \tag{4}$$

The mass conservation equation of condensate film:

$$\frac{\partial\rho_l h}{\partial t}+\nabla_s\cdot(\rho_l h\vec{V}_l)=\dot{m}_s \tag{5}$$

where h is the height of liquid film.

$$\frac{\partial\rho_l h\vec{V}_l}{\partial t}+\nabla_s\cdot(\rho_l h\vec{V}_l\vec{V}_l+\vec{D}_V)=-h\nabla_s P_L+\rho_l h\vec{g}_\tau+\frac{3}{2}\vec{\tau}_{fs}-\frac{3\mu_l}{h}\vec{V}+\dot{\vec{q}}_s+\vec{\tau}_{\theta_w} \tag{6}$$

The energy conservation equation of liquid film:

$$\frac{\partial}{\partial t}(\rho_v Y)+\nabla\cdot(\rho_v v Y)=\rho_v D\nabla Y+S_m \tag{7}$$

The mass flow rate of water steam in the interface was calculated using diffusion balance model:

$$\dot{m}_{phase}=\frac{(\rho D/\delta)}{\rho D/\delta+C_{phase}}C_{phase}(y_{sat}-y_i) \tag{8}$$

where $y_{sat}$ is the saturation mass fraction of water and can be calculated as follows:

$$y_{sat}=\frac{P_{sat}(T)}{P}\frac{M_i}{\bar{M}} \tag{9}$$

The momentum conservation of liquid film:

$$P_L=P_{gas}+P_h+P_\sigma \tag{10}$$

where

$$P_h=-\rho h(\vec{n}\cdot\vec{g}) \tag{11}$$

$$P_\sigma=-\sigma\nabla_s\cdot(\nabla_s h) \tag{12}$$

$$\frac{\partial\rho_l h T_f}{\partial t}+\nabla_s\cdot(\rho_l h T_f\vec{V}_l+\vec{D}_T)=\frac{1}{C_p}[\frac{2k_f}{h}(T_s+T_w-2T_m)+\dot{q}_{imp}+\dot{m}_{vap}L] \tag{13}$$

To compare 3-D finned tube with smooth tube, ratio of heat flux are defined as follows:

$$\varepsilon(\mathrm{Q}) = \frac{Q_{3\text{-}D\ finned\ tube}}{Q_{smooth\ tube}} \tag{14}$$

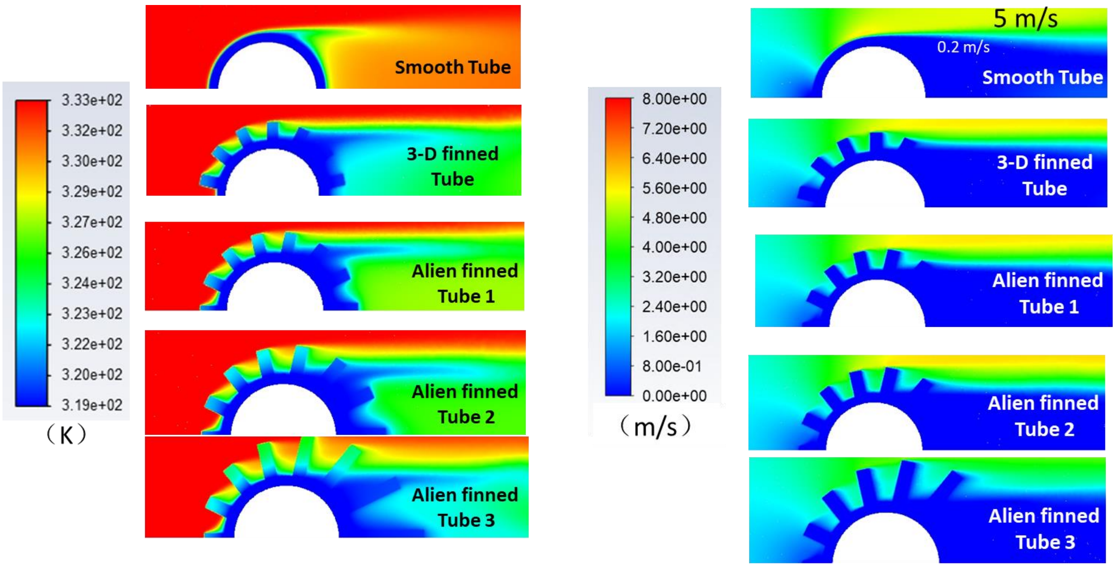
### 2.3 Result validation

Validation of numerical result is consisted of two parts. The first is grid independence verification. Mesh number varied from 5.5 M to 16.1 M were tested and 8.5 M was chosen considering accuracy and economy. Then, the results were validated against our experimental result carried out by Gu et al (2021). The deviation was within 30.1 % between experimental result and numerical result, which shows the result was reliable.

## 3. Results and discussion

### 3.1 Field distribution outside 3-D finned tube

The temperature and velocity was presented in Figure 2. In this section, the inlet temperature of moist air was 333 K. Inner tube wall temperature was 318.8 K. Inlet velocity was 2.4 m/s. Figure 2(a) shows that the temperature was low after heat transfer with smooth tube and 3-D fin. Besides, the increase of 3-D fin height is favourable for the turbulence of flow field in the tail of the flow around the tube. The main condensation occurs on the front of smooth tube and it mainly happens on the fin surface for 3-D finned tube. The rate of steam condensation lies on the thickness of steam concentration boundary layer. It develops with the flow outside smooth tube. However, the presence of 3-D fins can break the development of flow boundary, which is favourable for the condensation process. The phase change rate distribution was present in Figure 3. The results show that condensation rate was higher in the front of finned tube. The front of tube has almost two orders of magnitude higher than the rear. The reason is that the boundary layer is thinner on the fin and the front of base tube. Besides, the increment of 3-D fin height increased the condensate in the rear.




(a) Temperature field (b) Velocity field

*Figure 2: Distribution of velocity and temperature field*

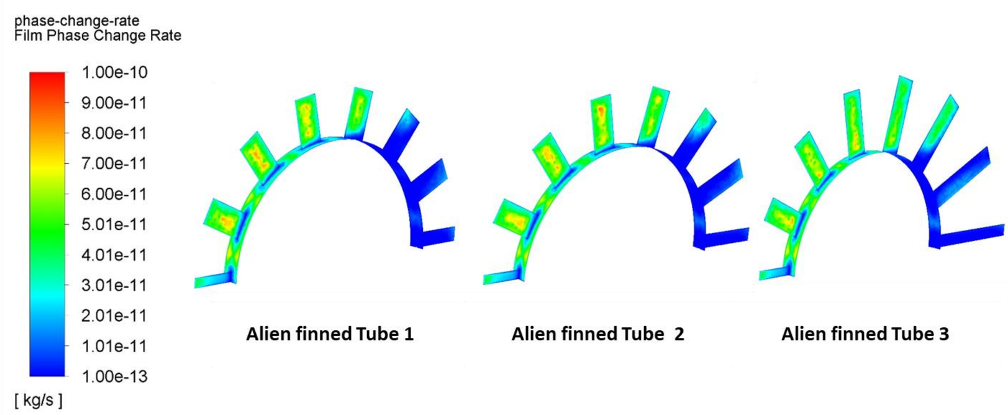



*Figure 3: Distribution of phase change rate*

### 3.2 Heat transfer performance

The heat flux of the tested tubes were presented in Figure 4. 3-D finned tubes has a higher heat transfer performance compared with smooth tube, and Alien finned tube 3 has the highest heat flux. The total heat flux ($Q_o$) was 4.39 times comparing with that of the smoot tube. The latent heat flux ($Q_L$) and sensible heat flux ($Q_S$) were 4.27 and 4.88 times, respectively. The area of Alien finned tube 3 was also the highest and was nearly 5 times that of smooth tube. Its enhancement of heat flux and heat transfer area was nearly the same. The result indicated that increase fin height in circumferential direction is an effective method to increase heat transfer.

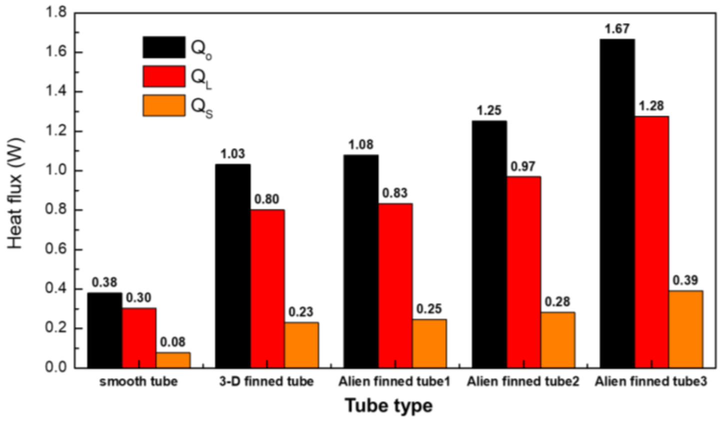



*Figure 4: Heat flux of smooth tube and 3-D finned tube*

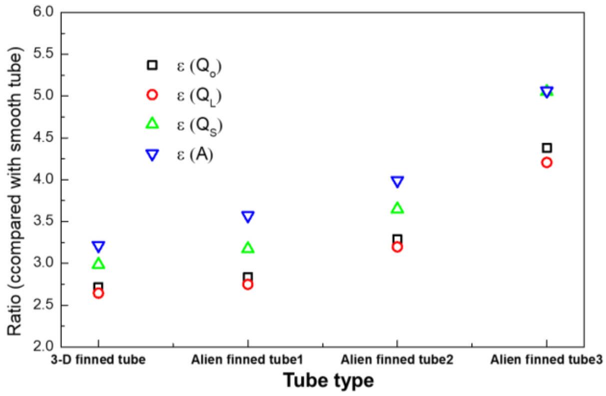



*Figure 5: Ratio of heat transfer between 3-D finned tube and smooth tube*

## 4. Conclusions

The condensation of moist air outside 3-D finned tubes with different circumferential fin height were numerically studied. Through comparing with smooth tube, the results indicated that adding 3-D fins made the main condensation place transferred from base tube to fin surface. Meanwhile, alien finned tube with highest fin height could achieve 4.27 times latent heat flux and 4.88 times sensible heat flux enhancement compared with smooth tube. Besides, increasing fin height in circumferential direction was favourable for the heat transfer enhancement in the trail area of tube, and sensible heat transfer coefficient was more sensitive than latent heat transfer. The results obtained in this study can give reference for the design of effective moist air condensation enhancement elements. In future, the influence of thermal conductivity and different moist air parameters should be considered.

### Nomenclature

h –fin height, m
$C_s$ – circumferential fin spacing, m
$C_{phase}$ – condensation coefficient
D – diffusion coefficient, $m^2/s$
F – fin number
v – velocity, m/s
t – time, s
T – temperature, K
S – source term
x, y – Cartesian coordinates
m – mass flow rate, kg/s
M – mole molecular weight
p – pressure, Pa
Q – heat flux, W
W – fin width, m
y – mass fraction
$\varepsilon$ – enhancement ratio
$\tau$ - shear stress
$\mu$ – dynamic viscosity, Pa s
$\rho$ – density, $kg/m^3$

### Acknowledgements

This study is supported by the Innovative Research Group Project of National Natural Science Foundation of China (No. 52021004). Scientific Research Found of Dongfang Turbine CO., LTD (2020-154600)

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