Heat transfer, erosion and acid condensation characteristics for novel H-type finned oval tube

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Abstract. Low efficiency of heat transfer, acid corrosion and erosion of economizers affect the economy and security in coal-fired power plants significantly. The H-type finned oval tube is proposed to alleviate these problems. Based on the H-type finned oval tube, we investigated three novel types of fins, including bleeding dimples, longitudinal vortex generators (LVGs), and compound dimple-LVG. We considered the three aspects together, and obtained the heat transfer, acid condensation rate and erosion loss. The results show that the tube bank with the new structured fins can improve the performance on the three aspects, and the compound dimple-LVG performs the highest comprehensive effect.

1. Introduction
The economizers in coal-fired boilers usually face three significant problems: low heat transfer efficiency of the flue gas, serious tube acid corrosion and erosion. Recently, many researchers have investigated one of the three aspects. Li et al. [1] studied the heat transfer and laminar flow characteristics of a slit finned tube heat exchanger with LVGs numerically. They also optimized the slit fin structure by using the field synergy principle. Bi et al. [2] numerically studied the convective heat transfer inside mini-channels with dimples, cylindrical grooves and low fins. They found that the dimple surface shows the highest performance of heat transfer enhancement. Wilson [3] and Zhang [4] simulated the acid condensation rate and water condensation rate of the laminar flow in a circular tube. The FLUENT solver with laminar model was employed to preliminarily simulate acid condensation rate on the water-cooled wall by Volz [5] and in the concrete chimney in power station by Jin et al. [6], respectively. They found that the acid condensation

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rate changes with the physical parameters. Ahmad et al. [7] experimentally studied the gas-liquid two-phase flow in heat exchanger. And they proposed a theoretical model to predict the convection-condensation and two-phase flow distribution. Nagarajan et al. [8] determined the effects of ash particle physical properties and transport dynamics on the erosive wear of three different grades of low alloy steel experimentally. Fan et al. [9-12] made a series of numerical calculation to predict the interaction between particles and tubes and the erosion. In these researches, some anti-wear structures were presented. Few researches, however, take all the three factors into account, and the comprehensive effect of the three aspects is more practical. In particular, for heat exchangers in the coal-fired power plants, it is necessary to consider not only the heat transfer and flow resistance performance, but also the acid anti-condensation and anti-wear performance. In this paper, we study the heat transfer, acid condensation and erosion characteristics of the single H-type finned oval tube, the H-type finned oval tube with bleeding dimples, with -30° rectangular longitudinal vortex generators (LVGs), and with compound dimples and LVGs. The purpose of this paper is to study the mechanism of heat transfer enhancement, acid condensation and erosion reduction of H-type finned oval tube with different vortex generators so that we can achieve a high efficient heat exchanger on heat transfer, acid anti-condensation and anti-wear.

2. Model description and numerical method

2.1. Physical model

A schematic diagram of a novel H-type finned oval tube with compound dimples and LVGs is displayed in Fig. 1. A pair of rectangle winglets is punched out from the fin symmetrically behind each oval tube and the bleeding dimples are penetrated out in the windward side. The geometry parameters of the H-type finned oval tube with dimples and -30° rectangular longitudinal vortex generators (LVGs) are listed in Table 1, and the oval tube minor radius and major radius are 40mm and 60mm, respectively.

![Fig. 1 Schematic diagram of a novel H-type finned oval tube with compound dimples and LVGs.](image)

2.2. Governing equations and boundary conditions

It is assumed that the gas-solid flow in the economizer is a three-dimensional, viscous and steady incompressible turbulent flow. The tube surface is assumed to be at a constant temperature $T_w=350$ K and the wall thickness is ignored due to the relatively large heat transfer coefficient between the cooling water and inner wall of the tubes and the high thermal conductivity of the tube wall. Moreover, the temperature distribution in the fin surface and in the gas fluid is determined by coupled computation. The solid fin and tube are assumed to have constant thermal conductivity. The condensation of water vapor and sulfuric acid vapor in the flue gas is negligible, but it is computed on the wall. The temperature of liquid film surface is
equal to the temperature of the wall. The convection caused by the diffusion is negligible.

Table 1 Geometry parameters of the basic H-type finned oval tube with dimples and -30° LVGs.

| $S_1$/mm | $S_2$/mm | $H_m$/mm | $W$/mm | $w$/mm | $F_i$/mm |
|----------|----------|-----------|--------|--------|----------|
| 140      | 100      | 5         | 26     | 72     | 50       |

| $F_1$/mm | $F_m$/mm | $F_p$/mm | $a_m$/mm | $b_m$/mm | $F_n$/mm |
|-----------|-----------|-----------|-----------|-----------|-----------|
| 3         | 40        | 15        | 5         | 10        | 36        |

| $r_1$/mm | $r_2$/mm | $r_3$/mm | $H$/mm | $m$/mm |
|----------|----------|----------|--------|--------|
| 1        | 0.4      | 1        | 95     | 12     |

2.2.1 Gas-phase governing equations

The gas-phase governing equations including mass, momentum, energy and composition conservation equations are as follows.

Continuity equation:

$$\frac{\partial}{\partial x_i}(\rho u_i) = 0$$

(1)

Momentum equation:

$$\frac{\partial}{\partial x_i}(\rho u_i u_j) = \frac{\partial}{\partial x_j}(\mu \frac{\partial u_i}{\partial x_j}) - \frac{\partial p}{\partial x_i}$$

(2)

Energy equation:

$$\frac{\partial}{\partial x_i}(\rho u_i T) = \frac{\partial}{\partial x_i}(\lambda \frac{\partial T}{\partial x_i}) + S_e$$

(3)

Where $\rho$ is the density, $u$ is the velocity, $T$ is the temperature, $\mu$ is the dynamic viscosity, $\lambda$ is the thermal conductivity, $C_p$ is the specific heat at constant pressure, and $S_e$ is the energy source term.

Species equation:

$$\nabla \cdot (\rho_i Y_i \mathbf{U}) = \nabla \cdot \left[ \rho D_{im} \left( \frac{\eta_i}{S_{C_i}} \right) \nabla Y_i \right] + S_{m}$$

(4)

Where $S_m$ is the species source term, $Y_i$ is the mass fraction of $i$ component, $S_{C_i}$ is the turbulent Schmidt number with $S_{C_i} = 0.7$, and $\eta_i$ is the turbulent viscosity. The Reynolds number of the flue gas flow in the present study ranges from 15677 to 24705, and the turbulence model is presented below.

The RNG $k$-$\varepsilon$ turbulence model:

$$\rho \frac{Dk}{Dt} = \frac{\partial}{\partial x_i} \left( \alpha_e \mu_{eff} \frac{\partial k}{\partial x_i} \right) + \mu S^2 - \rho e$$

(5)

$$\rho \frac{De}{Dt} = \frac{\partial}{\partial x_i} \left( \alpha_e \mu_{eff} \frac{\partial e}{\partial x_i} \right) + \frac{C_k}{k} S^2 - C \rho \frac{e^2}{k} - R$$

(6)

where $R$ is the rate of strain term given by
\[ R = \frac{C \phi^{3} (1 - \phi / \phi_{0}) e^{\varepsilon}}{1 + \beta \phi} \]  
(7)

where \( \phi = s k / \varepsilon \), \( \phi_{0} = 4.38 \), \( \beta = 0.012 \), \( C_{i} = 1.42 \) and \( C_{w} = 1.68 \). \( S^{2} = 2 S_{0} S_{g} \) is the modulus of the rate of strain tensor expressed as \( S_{ijk} = \frac{1}{2} (\partial u_{i} / \partial x_{j} + \partial u_{j} / \partial x_{i}) \).

2.2.2 Vapor condensation model

The mass condensation rate of the water and acid vapor at the wall surface is

\[ m_{j} = - \left( \rho D_{ij} + \frac{m_{j}}{S_{c_{i}}} \right) \nabla Y_{i} \]  
(8)

The effective diffusivity \( D_{ij} \) of \( i \) component should be calculated from the binary diffusivity coefficients of \( i \) component through other components in the flue gas. The binary diffusivities \( D_{ij} \) can be calculated using the Fuller correlation [3].

The acid solution concentration at the wall surface is

\[ y_{a} = \frac{R_{a}}{2 R_{a} + R_{w}} \]  
(9)

Where \( R_{a} \) and \( R_{w} \) are the mole condensation rate of the sulfuric acid vapor and the water vapor at the wall surface with \( R_{a} = m_{a} / M \), and \( M \) is the mole mass of sulfuric acid.

When the condensation occurs in the grid cells in the fluid region near the wall, the sulfuric acid vapor and water vapor concentration in the gas phase will decrease and the latent heat is released. So the species source and energy source in Eqs. (3) and (4) are redefined as

\[ S_{s} = -m_{a} A / V, \quad S_{m} = -m_{a} A / V \]  
(10)

Where \( A \) is the area of the grid on the condensing interface, \( V \) is the volume of the grid, and \( r_{i} \) is the latent heat of \( i \) component.

The partial pressure of the vapor at the gas-liquid interface is the key factor affecting the condensation rate. It can be calculated according to the gas-liquid equilibrium data [13].

\[ \ln p_{i} = a_{i} A + b_{i} B + c_{i} C + \Delta H_{i}^{v} D + \Delta S_{i}^{v} E + \tilde{C}_{i}^{m} F + \tilde{E}_{i}^{m} G + \alpha_{i} H + \ln \tilde{a}_{i} \]  
(298)

Where \( p_{i} \) is the monomer partial pressure at the gas-liquid interface, \( A-H \) are the equation coefficients that are affected by the temperature of acid condensate, \( a_{i} \), \( b_{i} \), \( c_{i} \), \( \Delta H_{i}^{v} \) and \( \Delta S_{i}^{v} \) are the thermodynamic parameters for pure components when the temperature is 298 K, and \( \tilde{C}_{i}^{m} \), \( \tilde{E}_{i}^{m} \) and \( \alpha_{i} \) are the parameters dependent on the composition change when the temperature is 298 K [14].

The hydration reaction of the sulfuric acid vapor and the water vapor lead to the production of \( H_{2}SO_{4}.H_{2}O \), so \( p_{i} \) is not the accurate partial pressure. Fugacity coefficient was introduced to adjust the vapor concentration. In the chemical theory of solutions, the apparent partial pressure is equal to the
monomer partial pressure divided by the apparent fugacity coefficient [15], given by
\[ p_{io} = p_i / \phi_{io} \]  
(12)

2.2.3 Lagrangian formulation for particle motion

The particles are assumed to be spherical points and we track each individual particle in the Lagrangian framework. In our assumption, the forces acting on the particles such as the gravity, thermophoretic force and Brownian force are ignored, but the Saffman lift [16] caused by the transverse velocity gradient is considered.

The force balance of the particles:
\[ \frac{du}{dt} = F_{D}(u - u_p) + \frac{g(\rho_p - \rho)}{\rho_p} + F_s \]  
(13)

where \( F_{D}(u - u_p) \) is the particle drag force per unit mass

\[ F_D = \frac{18\mu C_D Re_m}{\rho_p d_p^2} \frac{24}{24} \]  
(14)

\[ Re_m = \frac{\rho d_p |u_p - u|}{\mu} \]  
(15)

where \( u \) is fluid phase velocity, \( u_p \) is particle velocity, \( \mu \) is dynamic viscosity of the fluid, \( \rho \) and \( \rho_p \) are density of the fluid and particle, respectively, \( d_p \) is the particle diameter, \( C_D \) is the drag coefficient and \( Re_m \) is the relative particle Reynolds number.

2.2.4 Particle wall collision model and wall wear model

When fly ash particles collide with the surface of the tube and fin, the ash particles will change the trajectory as schematic in Fig. 2. To follow the particle trail, the collision model proposed by Tabakoff et al. [17] for coal ash particle impacting on stainless steel surface is employed. Normal and tangential coefficients of restitution can be assumed as constants during the collision of the particles with the tube and fin surface and presented as

\[ \frac{V_{n2}}{V_{n1}} = 1.0 - 0.4159 \beta - 0.4994 \beta^2 + 0.292 \beta^3 \]  
(16)

\[ \frac{V_{t2}}{V_{t1}} = 1.0 - 2.12 \beta + 3.0775 \beta^2 - 1.1 \beta^3 \]  
(17)

Fig. 2 Schematic of the particle collision with solid surface.

where \( V_n \) and \( V_t \) represent the particle velocity components normal and tangential to the tube surface, respectively. Subscripts 1 and 2 refer to the condition before and after collision, respectively. \( \beta \) is the angle
between the incident velocity and the tangent to the surface as schematic in Fig. 3. In the erosion model by Tabakoff et al. [17], the mass wear rate \( E \) is defined as the ratio of the amount of mass loss of the target metal material (milligram) to the collision particle mass (gram).

\[
E = K_1 (1 + C_k \left( \frac{90}{\beta_0} \right)) V_i^2 \cos^2 \beta (1 - R_i^2) + K_3 (V_i^2 \sin \beta)^3
\]  

(18)

where \( V_i \) and \( \beta \) indicate the particle impact velocity and collision angle, respectively. \( R_i = 0.0016 V_i \sin \beta \), and \( \beta_0 = 25^\circ \) is the angle when the maximum erosion occurs according to the experimental results of Tabakoff et al. [17]. \( C_k = 1 \) for \( \beta_1 \leq 3 \beta_0 \), while \( C_k = 0 \) for \( \beta_1 > 3 \beta_0 \). \( K_1, K_2 \), and \( K_3 \) are the empirical constants with \( K_1 = 1.505101 \times 10^{-6}, K_2 = 0.296007 \), and \( K_3 = 5.0 \times 10^{-12} \). Here, the employed tube material and the particles are the AISI 304 stainless steel and the anthracite, respectively.

2.2.5 Boundary conditions

The computational domain is composed of six boundaries: inlet, outlet, two symmetrical boundary (right and left), and two periodic boundary surfaces (top and bottom). To maintain the uniform inlet velocity and avoid the exit velocity with a recirculation-free flow, the computational domain is extended along the upstream by 3.5 times of the oval tube major radius length at the inlet zone and along the downstream by ten times of the oval tube major radius length at the exit zone, respectively. At the inlet boundary, the flue gas enters the computational domain along the \( x \) direction at uniform velocity \( u_{in} \), temperature \( T_{in} \) (450K), and the velocity components along \( y \) and \( z \) directions are set to be zero.

For the particle phase, the diameter distribution of particle group follows the Rosin-Rammler diameter distribution, which can be manifested that the maximum diameter, minimum diameter and mean diameter of particles are 0.03 mm, 0.0001 mm and 0.01 mm, respectively. One-way coupling is applied in the dilute dispersed two-phase flow, and the effect of the discrete phase on the continuous phase is ignored. In this paper, a three-dimensional and steady-state numerical simulation is conducted by using the commercial software FLUENT with a second-order discretization scheme for both convective and diffusive terms. When the residual of each variable for gas phase is below \( 10^{-4} \), the numerical computation is regarded as convergence. The erosion and sulfuric acid vapor condensation of tubes are measured by writing a UDF program. We deal with the mass transfer by solving the species equation and the source term in the equation.

2.3. Parameter definition

Some characteristic and non-dimensional parameters are defined as follows.

\[
Re = \frac{\rho u_{in} D}{\mu}, \quad h = \frac{Q}{\Delta T A_t}, \quad Nu = \frac{h D}{\lambda}, \quad Eu = \frac{2 \Delta p}{\rho u_{in}^2 Z}, \quad \Delta T = \frac{(T_{out} - T_{in}) - (T_{in} - T_{out})}{\ln((T_{out} - T_{in})/(T_{in} - T_{out}))}, \quad f = \frac{2 \Delta P}{\rho u_{in}^2 A_t},
\]

\[
\Delta P = P_{in} - P_{out}, \quad PEC = \frac{Nu}{Nu_0} \left( \frac{f}{f_0} \right)^{\frac{1}{3}}
\]  

(19)

In the above equations, \( u_{in} \) is average velocity in the minimum flow cross-section, \( Q \) is the heat transfer capacity, \( \Delta P \) is the total pressure drop over the whole computational domain, \( Z \) is the number of tube rows, \( A_m \) and \( A_t \) are the minimum flow cross-section area, and the total surface area, respectively. \( D \) is
the characteristic length, which is the length of the oval tube minor radius [18], and \( Nu_0 \) and \( f_0 \) are Nusselt number and friction factor of the compared baseline, respectively.

### 2.4 Grid generation and independence validation

The mesh is generated by using software GAMBIT 2.4.6 and the grid independence test is necessary to make sure the accuracy and validity of the numerical results. Here, we demonstrate the grid independence test of the tube rows with compound dimples and LVGs in detail. Three sets of grid numbers about 190 413, 367 937, and 887 451 cells are examined and the obtained corresponding \( Nu \) at \( Re = 15677 \) are 111.52, 113.04 and 115.02, respectively. The relative error of \( Nu \) between number 2 and number 3 is 1.7%, and the grid number of 113.04 is selected finally considering both the accuracy of the numerical results and the computer resource. Besides, similar tests of the grid independence are also performed for the other cases mentioned above. The detail results are not shown here to save the article’s space.

### 2.5 Validation of the computational model

Apart from the grid independence test, we should verify the computation accuracy of the model and calculation method. Here, the H-type finned tube bank is carried out for the validation. The simulated geometry parameters of the H-type finned oval tube are the same with the ones presented in Ref. [19]. The experimental correlations from Ref. [19] for the Nusselt number and Euler number are presented as

\[
Nu = 0.09152Re^{0.7015}Pr^{0.33} \quad \text{and} \quad Eu = 0.2963Re^{-0.449},
\]

respectively. The comparison of numerical results with the corresponding experimental correlations is displayed in Fig. 3. The average discrepancy and maximum discrepancy between the predicted \( Nu \) and the experimental correlation are 5.0% and 5.2%, respectively, and the average discrepancy and maximum discrepancy between the predicted \( Eu \) and the experimental correlation are 8.9% and 10.8%, respectively. However, there are still some obvious deviations between the numerical results and the solutions from experimental correlations, which may be ascribed to the system error in the experiment and the assumptions in the numerical simulation. These deviations could be allowable, and therefore, we can believe that the present numerical method meets the computational requirements for engineering applications.

![Fig. 3 Model validation with experimental data.](image)

### 3 Numerical results and discussion

In the following sections, the longitudinal vortices generated by the rectangular winglet and dimples are presented. Then the heat transfer, acid anti-condensation and anti-wear characteristics based on the single H-type finned oval tube with different enhanced structures are discussed.
3.1. The influence of rectangular winglet and dimples on flow field

Fig. 4 presents the streamlines starting near dimples for \( u = 7 \text{ m} \cdot \text{s}^{-1} \) at \( x = -26 \text{ mm} \). When the gas flows over the dimples, the gas develops cyclones inside the dimples as shown in Fig. 4 (b). Zhang et al. [20] pointed that the pits in the surface could decrease both the turbulence intensity near the wall surface and the friction resistance through lowering the velocity gradient. That is because the low speed cyclones inside the pits lead to interaction between the inner flow and the outer flow of the dimples called air cushion, which is the fluid control behavior on the wall boundary layer developed by dimples surface. The friction drag generated from the bottom of dimples acts as an accessional impetus. When flowing over the LVG, the air in the back surface flows to the front fin surface because the pressure in the front surface is lower than that in the back surface. The streamlines traversing across the fin through the bleeding hole and the LVG enhance strongly the fluid transport between the mainstream region and the wake region, which can be observed from Figs. 4 (a)-(c). Fig. 4 (d) shows the velocity vectors of the cross-section at \( x = 50 \text{ mm} \) behind the fin surface. It can be observed that the vortex is formed behind the fin, which is usually defined as the secondary flow. The structures of secondary flow generated by the LVG have been sufficiently studied and more details can be found in Ref. [21].

![Streamlines starting from the single tube near dimples in (a), (b) and (c), and velocity vectors of the cross-section at \( x = 50 \text{ mm} \) behind the fin in (d).](image)

3.2. Flow and heat transfer characteristics of single tube

Fig. 5 illustrates the local velocity distribution of the \( XY \)-plane in the middle-plane \( (Z=0) \) at \( u = 7 \text{ m} \cdot \text{s}^{-1} \). It can be seen that behind the tube, because of the obstruction of the tube, the gas separates into two regions, the mainstream region and the wake region. As can be observed from Fig. 5 (a), the flow velocity is very low in the rear of the tube (the wake region). Behind the tube, we can clearly see from the streamlines distribution that the vortices are generated, which are called transverse vortices. The existence of the
transverse vortices shows the recirculating of the fluid in the wake region, where the fluid is almost separated from the main flow. The effect of dimples on the velocity field can be observed from Fig. 5 (c). Compared with Fig. 5 (a), the wake region in Fig. 5 (c) is compressed and the streamlines in the centers of the transverse vortices become dense. Besides, comparing with Figs. 5 (b) and (d) with Fig. 5 (a), we can see that the wake region is reduced when the gas flows over the LVG. The LVG is able to increase the flow transport between the mainstream region and wake region so that enhance the flow of wake region and delay the flow separation behind the tube.

![Fig. 5](image_url)

**Fig. 5** Local velocity distribution on the middle cross-section in x direction. (a) H-type. (b) LVG. (c) Dimple. (d) Compound dimple-LVG.

Fig. 6 presents the Nusselt number $Nu$ versus Reynolds number in which both the Nusselt number and pressure drop for all the cases increases with the increasing Reynolds number. The Nusselt numbers of the finned oval tubes with different heat transfer enhancement structures are significantly higher than the original H-type finned oval tube. Among the three types of enhanced structures, the dimple, the LVG and the compound dimple-LVG, the enhanced heat transfer of the compound case achieves the highest, followed by the LVG case and the dimple case. Compared to the baseline, the increase of $Nu$ for the three types of enhancement structures from low to high is 25.7-34.8%, 36.4-46.3%, and 44.7-49.9%, respectively.

![Fig. 6](image_url)

**Fig. 6** Nusselt number and pressure drop versus Reynolds number

Fig. 7 shows the relationship between the friction factor and Reynolds number. It can be seen that the friction factor of the H-type finned oval tube with different augment structures are obviously higher than the original H-type finned oval tube. In addition, the friction factor from low to high is the dimple, the LVG, and the compound dimple-LVG cases, and the corresponding friction factor increase compared with
the H-type finned oval tube is 34.7-37.5%, 48.5-51.8%, and 48.2-60.2%, respectively. However, at low Reynolds number, the friction factors for the tube with LVG and the tube with dimple-LVG are very close. The results suggest that the dimple could slow the speed of the friction increase at low Reynolds number, and it is in consistent with the conclusion in Ref. [20].

Fig. 7 Friction factor versus Reynolds number

Fig. 8 presents the parameter of PEC for the finned oval tube with different enhanced structures against Reynolds number. The PEC is a universal evaluation parameter which represents comprehensive performance of a heat transfer unit. The H-type finned oval tube is regarded as the compared baseline. From Fig. 10 we can see that the value of PEC from high to low is the compound dimple-LVG, the LVG, and the dimple. The compound dimple-LVG provides higher overall heat transfer performance than the LVG or the dimple, which is resulted from the hybrid effect. These results show that the tube with compound dimple-LVG case achieves considerable augmentation of heat transfer capacity with moderate flow resistance increase.

Fig. 8 PEC versus Reynolds number

3.3. Sulfuric acid vapor condensation characteristics of single tube

From Fig. 9, we can find that the distribution of the acid solution concentration is similar to that of the temperature. The high concentration of the acid solution distributes most on the fin surface instead of the tube wall. Due to the cooling of the tube, the acid solution concentration decreases first and then increases along the streamwise direction of the flue gas. Therefore, we can make sure that the fin surface can protect the tube from corrosion significantly. In addition, the acid condensation rate is higher in the central area of the fin and becomes lower at the rear of the fin close to the tube wall. This is because the larger velocity of
flue gas in this zone can enhance the mass transfer between the acid vapor in the gas phase and the acid vapor at the gas-liquid interface. We can also find that both the dimples and LVGs have important effect on the acid condensation rate.

In Figs. 10, the inlet velocity is $u_{in}=8 \text{ m} \cdot \text{s}^{-1}$ and the temperature of the tube wall is $T_w=350 \text{ K}$. We can see that the acid condensation rate achieves the lowest for the finned surface with only dimples while the highest for the finned surface with only LVGs. Because the low velocity of the fluid near the dimple reduces the mass transfer between the acid vapor in gas phase and the acid vapor at the gas-liquid interface, which will lead to decrease of the acid condensation rate. The LVG structure can destroy the fluid boundary layer, which leads to flow disturbance and reduces the mass transfer resistance, so the LVG structure increases the acid condensation rate in the fin wake zone. It is interesting that in Figs. 10 (a), (b) and (c), the acid condensation rate is all about 1.4% higher for the LVG while about 6.3% and 4.7% lower for the Dimple and the compound Dimple-LVG, compared to the baseline of the H-type finned oval tube. Because the different finned tubes change the acid condensation rate mainly by breaking the flow state of the flue gas, however, the difference in the flow state of the flue gas is negligible among various fin structures at the present range of physical parameters ($T_{in}$, $x_a$ and $x_w$).

**Fig. 9** Contours of temperature, acid solution concentration and acid condensation rate on different fin surfaces.
Fig. 10 Effects of physical parameters on acid condensation rate for different finned tubes. (a) Temperature. (b) Sulfuric acid vapor concentration. (c) Water vapor concentration.

3.4. Erosion characteristics of single tube

Figure 11 shows the erosion contours of the solid at \( u = 7 \text{ m/s} \). We can see that the fin erosion is mainly distributed on both verges of the fin, which can be explained that those zones have large flow speed and the erosion is proportional to the cube of particle speed. Because the augment structures, namely the dimples or LVGs, change the fluid field, the erosion loss of the solid tube and fin is decreased compared with the original H-type finned oval tube. Moreover, it can be seen that the maximum erosion of
oval tube is concentrated in the range of 20-45° in windward of the oval tube.

![Erosion Contours](image)

**Fig. 11** Erosion contours of solid wall and fin. (a) H-type. (b) LVG. (c) Dimple. (d) Compound dimple-LVG.

From Fig. 12, we know that the tube wall erosion loss for H-type finned oval tube without any enhancement technique is the largest under the same Reynolds number. Specifically, compared with the H-type finned oval tube without any enhancement technique, the tube wall erosion loss of the compound dimple-LVG case and the dimple case is almost the same and reduced by 46.1-49.1%. The tube wall erosion loss for the H-type finned tube with rectangular LVG is reduced by 76.4-76.9%. Furthermore, the anti-wear performance of the enhanced tubes improves as the inlet velocity increases.

![Tube Erosion Value vs. Reynolds Number](image)

**Fig. 12** Tube erosion value versus Reynolds number.

4. Conclusions

In this study, the heat transfer, sulfuric acid vapor condensation and erosion characteristics of the H-type finned oval tube, tube with LVGs, tube with bleeding dimples, and tube with compound dimples and LVGs have been numerically studied. The following conclusions are obtained.

1. The comprehensive performance of the enhanced heat transfer surfaces is evaluated by the parameter PEC. The H-type finned oval tube with compound dimples and LVGs provides the highest overall heat transfer performance. The performance of the tube with LVG is lower than that of the tube with compound dimple-LVG, and the tube with bleeding dimple presents the lowest performance.

2. Both the finned tube with bleeding dimples and the finned tube with compound dimples and LVGs can inhibit acid vapor condensation effectively because the acid condensation rates of them are lower than...
that of the H-type baseline under the same conditions. Furthermore, the finned tube with bleeding dimples performs slightly higher.

3. The anti-wear performance of the tube with LVGs is the highest, and it is obviously higher than that of the tube with compound dimple-LVG and with the sole dimples. For all the enhanced structures, the erosion loss of the tubes are dramatically less than the original H-type oval tube, and the reduction of erosion loss gets more for a larger Reynolds number.

The H-type finned oval tube with compound dimple-LVG heat exchanger may have significant applications not only to improve the heat transfer efficiency and economy, but also reduce acid condensation and the wear loss of heat exchanger and promote the safe operation level in coal-fired boilers.

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