Research On Waste Heat Recovery Of Low Temperature Flue Gas In Cement Plant And New Type Heat Exchange

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ABSTRACT

Based on the measured data of low-temperature waste gas at the outlet of waste heat boiler of a new dry cement production line in a cement factory in China, this paper investigates the utilization of waste heat, and formulates the technical scheme of waste heat utilization, and focuses on the design of the key waste heat exchanger for waste heat utilization, and carries out technical and economic analysis. According to our calculation results, the low-temperature waste heat recovery of cement plant has great potential. Using this technology can save more energy and obtain good economic benefits.In addition, a new type of heat pipe heat exchanger is designed and briefly introduced in this paper, aiming at the current situation that the existing heat exchange equipment only has heat exchange function without considering dust removal function.The design inspiration of the new type of heat pipe heat exchanger comes from cyclone and heat pipe heat exchanger, which combines the advantages of high heat exchange efficiency of heat pipe heat exchanger and good dust removal effect of cyclone.Heat exchange equipment for low temperature waste heat recovery needs to have some unique performance requirements.It can provide better choice for waste heat utilization of cement plant.

1. INTRODUCTION

The heat loss of cement plant mainly includes radiation heat loss of kiln shell, heat loss of waste gas and fly ash, heat loss of slag, etc (Doheim *et al.*,1987). At present, the waste heat exchange of cement kiln waste gas generates low-pressure superheated steam through waste heat boiler, which drives the steam turbine generator set to generate electricity.The technology has been developed and mature in China. However, the temperature of the low-temperature exhaust gas discharged from the outlet of the waste heat boiler is still 100-200 ℃, the total flow of this part of the low-temperature exhaust gas is large, and the potential of the recoverable waste heat is huge. If the low-temperature exhaust gas continues to cool down to 50-80 ℃ by developing and using the low-temperature waste heat recovery technology, the huge economic, environmental and social benefits will be obtained.Tchanche *et al.* (2010) studied the solution of using organic Rankine cycle (ORC) to recover waste heat from thermal process for power generation.Y. *et al.* (2017) used multi-stage flue gas cooler to recover waste heat boiler of power plant, analyzed the deep utilization of flue gas waste heat, and proved that the unit has significant economic benefits in heating period and non heating period. The development and research on the related low-temperature waste heat recovery equipment and the ash cleaning and wear prevention technology of the equipment will also be affected to more and more experts, scholars and technicians.

2. DESIGN CALCULATION OF HEAT EXCHANGER

Table 1: Parameters of three pipelines in the first line of the kiln

	Temperature（℃）	Pressure（Pa）	Working air volume（m³/h）	Standard condition air volume（Nm³/h）
1	100	4900	49900	38176
2	100	4500	45900	34983
3	100	3700	76300	57712

2.1 Design parameters

Flue gas inlet temperature: T_1' = 100℃;flue gas outlet temperature : T_1'' = 60℃;flue gas mass flow: M_1 = 123803.97kg / h; specific heat capacity at constant pressure of flue gas: C_{p1} = 1.009kJ / (kg·℃); water inlet temperature: T_2' = 20℃; water outlet temperature: T_2'' = 55℃. Therefore, the total heat of the flue gas:

$$Q_0 = M_1 C_{p1} T_1' = \frac{123803.97 \times 1.009 \times 100}{3600} = 3469.95\text{k}W \tag{1}$$

Recovery of residual heat:

$$Q_1 = M_1 C_{p1} \left(T_1' - T_1''\right) = \frac{123803.97 \times 1.009 \times (100 - 60)}{3600} = 1387.98\text{k}W \tag{2}$$

Waste heat recovery rate:

$$Q_1 / Q_0 = 40\% \tag{3}$$

The calculation of the amount of hot water M_2 that the heat pipe heat exchanger can heat, according to the heat balance equation:

$$M_1 C_{p1} \left(T_1' - T_1''\right) = M_2 C_{p2} \left(T_2'' - T_2'\right) \tag{4}$$

$$M_2 = 34203.08\text{kg/h}$$

2.2 Basic selection of heat pipe heat exchanger

(1) Selection of working medium

①Temperature estimation in the heat pipe at the flue gas inlet:

$$T_v = \frac{T_1' + nT_2''}{1+n} \tag{5}$$

$$n = \frac{K_2 A_2}{K_1 A_1} \tag{6}$$

Where n is proportional constant. K_1 and K_2 are the heat transfer coefficients of the evaporation section and the condensation section, and A_1 and A_2 are the external surface areas of the evaporation section and the condensation section.

For gas-liquid heat pipe heat exchangers, when the heat exchange is water, n is usually 3 ~ 4.Take n = 3 here.

$$T_{v1}=\frac{100+3\times 55}{1+3}=66.25℃$$

②Estimation of temperature inside the pipe at the flue gas outlet:

$$T_v=\frac{T_1^{''}+nT_2^{'}}{1+n} \tag{7}$$

$$T_{v2}=\frac{60+3\times 20}{1+3}=30\ ℃$$

According to the inlet temperature, water is selected as the working medium of the heat pipe.

(2) Pipe selection

Considering the compatibility of the heat pipe wall with the working medium in the pipe and the economic efficiency of the heat pipe heat exchanger, a carbon steel coated pipe is used here as the heat pipe. The coating on the inner surface of the carbon steel pipe is due to the incompatibility between carbon steel and water. The reaction will generate a small amount of air bubbles and affect the heat transfer effect of the heat pipe.

(3) Choice of placement form and core structure

It is installed on the horizontal flue and adopts gravity heat pipe without liquid wick structure.

(4) Choice of pipe diameter and extended surface

①Choice of pipe diameter

The required pipe diameter determined by the speed of sound limit is:

$$d_v=1.54\sqrt{\frac{Q_C}{r(\rho_v P_v)^{\frac{1}{2}}}} \tag{8}$$

In the formula (7): d_v is the diameter of the cross section of the steam circulation in the tube (mm); Q_c is heat transfer heat at the speed of sound limit (kW);r is latent heat of vaporization (kJ / kg); ρ_v is density of steam in the tube (kg / m^3); P_v is the pressure of steam in the tube (N / m^2).

Take Q_c = 6kW, when T_v = 50 ℃, ρ_v = 0.083kg / m^3, P_v = 0.123×105N / m^2, r = 2383kJ / kg.

$$d_v=1.54\sqrt{\frac{6}{2383\times(0.083\times 0.123\times 0.123\times 10^5)^{\frac{1}{2}}}}=0.0137m=13.7mm$$

The required pipe diameter determined by the carrying limit is ：

$$d_v'=\sqrt{\frac{1.28Q_{ent}}{\pi\cdot r\cdot\left(\rho_l^{\frac{-1}{4}}+\rho^{\frac{-1}{4}}{}_v\right)^{-2}\left[g\sigma(\rho_l-\rho_v)\right]^{\frac{1}{4}}}} \tag{9}$$

In the formula (8): Q_{ent} is carrying the limit heat transfer heat (kW);r is latent heat of vaporization (kJ / kg); ρ_v is the density of steam in the tube (kg / m^3); ρ_l is the density of liquid in the tube (kg / m^3); σ is surface tension (N / m); g is gravitational acceleration (m / s^2).

Take Q_{ent} = 6KW, when the working temperature T_v = 60 ℃ in the tube, ρ_l = 983.2kg / m³, ρ_v= 0.1302kg / m³, σ = 662.2 N / m. So d_v' = 0.031m = 31mm.

Taking into account the design margin and for safety reasons, the final selected pipe diameter is $d_i=35$mm.

② Calculation of pipe wall thickness[2]:

$$S=\frac{P_V d_i}{200[\sigma]} \tag{10}$$

Where S is wall thickness (mm); [σ] is allowable stress (Pa); P_v is allowable pressure in the pipe (Pa).

Among them, P_v is selected according to the allowable pressure of water-steel heat pipe 40kgf / cm^2. When the temperature is 250 ℃, the maximum stress of the shell at this time is σ_{max} = 14kgf / mm^2, and the allowable stress is [σ]=1/4σ_{max}=3.5kgf/mm^2.

Here $S=\frac{40\times 35}{200\times 3.5}=2$mm

Considering the safety of the designed heat exchanger, take S = 2.5mm.

The outer diameter of the heat pipe is d_0=d_i+2S=39mm, So choose d_0=40mm.

The flue gas side uses finned tubes, the finned tubes are spiral fins, the thickness of the fins is 1mm, and the water side uses light tubes. The specific parameters of the heat pipes are shown in **Table 2**:

Table 2: Structure size of a single heat pipe

Light pipe outer diameter d_0(mm)	Light pipe inner diameter d_i(mm)	Fin outer diameter d_f(mm)	Fin height H(mm)	Fin thickness δ(mm)	Fin gap Y(mm)	Wing ratio β
40	35	72	16	1.0	8	6.24

The calculation of winging ratio is as follows：
The number of fins on a 1m tube is:1000/9 = 111.1 pieces.
The fin area on a 1m tube length is:

$$A_f = 111.1 \times \left[\frac{\pi}{4} \left(d_f^2 - d_0^2 \right) \times 2 + \pi \cdot d_f \cdot \delta \right] = 0.6504 m^2 \tag{11}$$

The bare tube area between the fins on a 1m tube length is:

$$A_b = \pi d_0 Y \times 111.1 = 0.1535 m^2 \tag{12}$$

The area of the light pipe at 1m length is:

$$A_0 = \pi d_0 \times 1 = \pi \times 0.04 = 0.1256 m^2 \tag{13}$$

The wing ratio is:

$$\beta = \frac{A_f + A_b}{A_0} = \frac{0.6504 + 0.1256}{0.1256} = 6.24 \tag{14}$$

2.3 Estimation and structural design of heat pipe heat exchanger

(1) Selection of inlet mass flow rate
Take the mass flow velocity of the inlet of the flue gas side: G_1=4 kg/m^2·s
The inlet mass flow of water intake: $G_2 = \rho_1 \cdot v_1 = 150$ kg/m^2·s
Here the water flow velocity v_l= 0.15m / s, to ensure that it is not less than 0.1m / s at the narrowest cross section.
(2) Selection of length ratio of heating section and cooling section
Calculate the economic length ratio

$$L_j = \sqrt{\frac{K_2}{K_1}}$$

where K_2 is taken as 2500W/（m^2·°C） and K_1 is taken as K_1 = 40β = 40 × 6.24 = 249.6W/（m^2·°C）, so L_j= 3.2.
The safe length ratio is calculated by:

$$[L] = \frac{K_2}{K_1} \times \frac{[T_v] - T_2}{T_1 - [T_v]} \tag{15}$$

In the formula (14): K_1, K_2 is heat transfer coefficient of heating section and cooling section; T_1,T_2 is temperature of hot fluid and cold fluid (°C). [T_v] is the working fluid saturation temperature (°C) corresponding to the maximum pressure allowed in the heat pipe.
Since the allowable pressure of the water-steel heat pipe is 40kgf / cm^2, that is, 40 × 10^5Pa, check the thermophysical property table of the saturated water to see that the corresponding saturated water temperature is 250°C and the temperature of the flue gas inlet is 100°C<250°C, so the heat pipe is safe to work at this temperature, there is no problem of safe length ratio.
(3)The windward area of the flue gas side and the heat transfer area of the first row of pipes.
Windward area: A_w=M_1/G_1=123803.97/4×3600=8.60m^2
If the width of the windward side is selected to be 3.6m, the length of the heat pipe is as follows:
Length of heating section: L_1 = 2.4m
Cooling section length: L_2 = 0.8m
Total length: L = L_1 + L_2 = 3.2m
Actual length ratio: L_1 / L_2 = 3
Actual windward area: A_{aw} = 3.6 × 2.4 = 8.64m^2
Actual mass flow at the gas inlet:
G_{am}=M_1/A_{aw}=123803.97/(3.6×2.4×3600)=3.98kg/(m^2·s)
Select the tube spacing S_1 = 80mm, then the number of tubes in the first row is: n = 45
The heat transfer area of the first row of tubes is: $A_{01} = \pi d_0 L_1 n_1 = \pi \times 0.04 \times 2.4 \times 45 = 13.56$m^2

(4)Selection of heat transfer coefficient K_0:

$$K_0 = \left[\frac{1}{\frac{1}{\alpha_1 \cdot \beta} + \frac{1}{\alpha_2} \cdot \left(\frac{L_1}{L_2}\right)}\right] \times 0.8 \tag{16}$$

In the formula (15): α_1 is the heat dissipation coefficient of the gas based on the surface area of the finned tube, take α_1 = 60W /（m²·°C）; β is wing ratio; α_2 is the heat transfer coefficient from the outside to the inside of the tube on the water side, take α_2 = 3000W /（m²·°C）; L_1/L_2 is length ratio, take 3; 0.8 is corrected value considering the influence of other thermal resistances.

$$K_0 = \left[\frac{1}{\frac{1}{60 \times 6.24} + \frac{1}{3000} \times 3}\right] \times 0.8 = 217.93\text{W} / (\text{m}^2 \cdot ℃)$$

(5) Calculate the logarithmic average temperature difference
Smoke:100°C→60°C
Water:55°C←20°C
Therefore, it can be calculated by the following formula:

$$\Delta T = \frac{(\Delta T)_{\max} - (\Delta T)_{\min}}{\ln \frac{(\Delta T)_{\max}}{(\Delta T)_{\min}}} \tag{17}$$

$$\Delta T = \frac{45 - 40}{\ln \frac{45}{40}} = 42.25$$

Estimated heat transfer area:

$$A_0 = \frac{Q_1}{K_0 \Delta T} = \frac{1387.98 \times 103}{217.93 \times 42.45} = 150.03\text{m}^2 \tag{18}$$

(6) Number of tubes
Total number of pipes:

$$n_p = \frac{A_0}{\pi d_0 L_1} = \frac{150.03}{\pi \times 0.04 \times 2.4} = 497.72 \tag{19}$$

Considering the design margin, select n_p= 540.
The number of tube rows is arranged in a fork row, with 45 in each row, and the total number of rows is N = 540/45 = 12 rows
The actual heat transfer area is:

$$A_{at} = \pi d_0 L_1 \times 540 = 162.78m^2 \tag{20}$$

The heat transfer of each heat pipe is:

$$q = \frac{Q}{540} = \frac{1387.98}{540} = 2.57kW \tag{21}$$

Selection of Water Side Pipeline
The required circulation area of each tube is:

$$A_{re} = \frac{M_2}{G_2 \times 3600} = \frac{34203.08}{150 \times 3600} = 0.063m^2 \tag{22}$$

Each row of pipes is selected as a pipe for water flow, that is, it is divided into 12 pipes on the water side, and the flow area of each pipe is: A'_{re}= 0.8 × 0.08 = 0.064 ㎡
The actual mass flow rate of water is:

$$G_{a2} = \frac{34203.08}{0.064 \times 3600} = 148.51kg/(\text{m}^2 \cdot \text{s}) \tag{23}$$

2.4 Resistance calculation of heat pipe heat exchanger

The smoke side resistance is calculated by the following formula (24):
In the formula (23): N is the number of pipe rows in the flow direction; ρ is fume density;g is gravity acceleration; G_{max} is the mass flow rate of flue gas at the narrowest section; f is coefficient of friction.
Here in formula (25)

$$\Delta P = f\frac{N\cdot G^2{}_{max}}{g\cdot \rho} = 0.513\times\frac{12\times7.96^2}{9.8\times1} = 39.8mmH_2O \approx 398Pa \tag{24}$$

$$f = 18.93\left(\frac{d_0 G_{max}}{\mu_1}\right)^{-0.316}\left(\frac{S_1}{d_0}\right)^{-0.927}\left(\frac{S_1}{S_2}\right)^{0.515} = 0.513 \tag{25}$$

In the formula (24): S_2 is Length of the hypotenuse of the tube bundle triangle arrangement , and $S_2=S_1 \sin60°$.

Then the resistance of each row of tubes is : ΔP/N = 398/12 = 33.2 Pa.

2.5 Calculation of insulation structure of heat pipe heat exchanger

The heat insulation of the heat exchanger adopts the method of wrapping the outer surface of the heat exchanger with rock wool board. The thermal conductivity of the rock wool board is 0.042W / (m •K), and the thickness of the rock wool board is 20mm. In the calculation, taking the meteorological parameters of the city where the cement plant is located as an example, the annual average wind speed is 3.4m / s and the annual average temperature is 13℃.

(1) Heat loss of heat exchanger

$$\Delta Q = \frac{(t-t_0)l}{\frac{1}{2\pi\lambda_b}\ln\frac{d_z}{d_w}+\frac{1}{\pi d_z\alpha_w}} = \frac{(100-13)\times3.2}{\frac{1}{2\pi\times0.042}\ln\frac{0.08}{0.04}+\frac{1}{\pi\times0.08\times24.5}} = 99.8W \tag{26}$$

In the formula (25): t is flue gas temperature in heat exchanger (℃);t_0 is the ambient temperature around the heat exchanger, take t_0 = 13℃; λ_b is the thermal conductivity of insulation material, take λ_b= 0.042W / (m. ℃); d_z is the diameter of the outer surface of the heat exchanger after adding the insulation layer (m); d_w is outer diameter of heat exchanger (m); a_w is the heat release coefficient of the outer surface of the insulation layer to the air.

a_w= 11.6 +7$\sqrt{v}$ ℃ , v is the wind speed; l is height of heat exchanger (m).

(2) The outer surface temperature of the heat insulation layer of the heat exchanger

$$t_s = \frac{\frac{1}{\lambda_b}\ln\frac{d_z}{d_w}t_0+\frac{2000}{\alpha_w d_z}t}{\frac{1}{\lambda_b}\ln\frac{d_z}{d_w}+\frac{2000}{\alpha_w d_z}} = 24℃ \tag{27}$$

(3) The ratio of heat loss on the surface of the heat exchanger to the heat recovery

$$\frac{\Delta Q}{Q_1} = \frac{0.0998KW}{1387.98KW} = 0.007\% \tag{28}$$

3. DESIGN OF NEW LOW-TEMPERATURE EXHAUST HEAT EXCHANGER

The design purpose of this new type of heat pipe heat exchanger is to develop a new type of equipment suitable for both low temperature waste gas waste heat recovery in the cement plant that has both heat exchange and dust removal functions.The design inspiration of the new heat pipe heat exchanger comes from cyclone dust collector and heat pipe heat exchanger, which combines the advantages of high heat exchange efficiency of heat pipe heat exchanger and good dust removal effect of cyclone dust collector. The initial idea is as follows:

(1)The new heat pipe heat exchanger has a cylindrical structure, and the inside is an independent heat pipe. The heat exchanger is divided into a gas side and a water side by a partition inside the heat exchanger. In the condensation section of the heat pipe, the flue gas flows through the evaporation section and transfers heat to the low-temperature water in the condensation section through the heat pipe.

(2)The flue gas side and the water side inside the heat exchanger are separated by a partition into multiple flow channels to improve the heat exchange effect, wherein the role of the flue gas side partition is to make the flue gas form a cyclone dust after entering the heat exchanger The spiral flow channel inside the device, the dust particles in the flue gas are thrown to the inner wall surface of the heat exchanger under the action of centrifugal force, and fall under the action of gravity; the role of the water side partition is to increase the flow rate of water Increase the heat transfer coefficient between water and heat pipe.

(3)In the evaporation section of the heat pipe, a spiral fin tube is added to the heat pipe. The spiral fin tube not only enhances the heat transfer effect of the flue gas side, but the downward tilt angle of the spiral fin is conducive to the accumulation of ash deposited on the fin surface. Fall under the influence of gravity.

(4)An ash discharge hopper is provided at the bottom of the heat exchanger, and the ash accumulation falling from the inner wall surface of the heat exchanger and the spiral fin tube can be conveniently discharged from the ash hopper.

Based on the above design ideas, the cross-sectional view of the new heat pipe heat exchanger is shown in Figure 4-1. The new heat pipe heat exchanger is equipped with 150 heat pipes, the length of a single heat pipe is 2.5m, and the length of the evaporation section is 2m The length of the condensing section is 0.5m.The heat pipes are installed in the space between the inner cylinder and the outer cylinder of the heat exchanger. There are 10 heat pipes in a row with a total of 15 rows.The flue gas enters from the tangential direction of the outer cylinder and rotates around the flue gas flow path between the inner cylinder and the outer cylinder. This process can make the dust contained in the flue gas be separated under the action of centrifugal force during the rotation to the inner wall surface of the outer cylinder, when the dust accumulates to a certain thickness on the inner wall surface, it can fall into the ash discharge hopper at the bottom of the heat exchanger under its own gravity.The flue gas rotates around the inner cylinder once and reaches the baffle. The baffle on the flue gas side of the heat exchanger has a certain arc to make the flue gas easily flow into the inner cylinder and rise in the inner cylinder. The flue gas outlet at the top of the heater is discharged.The water side of the new heat pipe heat exchanger is divided into 10 flow channels by a partition, the width of a single flow channel is 80mm, the water side heat tube is a smooth tube, and no fins are added.The low-temperature water flows into the heat exchanger from the middle inlet on the water side, rotates from inside to outside along the flow path inside the heat pipe heat exchanger, and finally flows out from the outer water outlet on the water side.To prevent the low-temperature water on the water side from generating steam during heating and causing excessive pressure in the cavity on the water side of the heat exchanger, a safety valve that opens automatically is provided at the top of the heat exchanger.

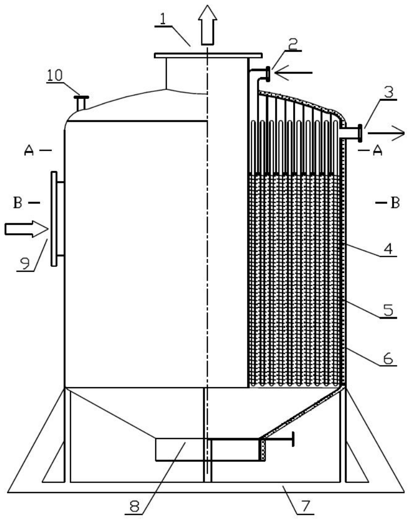
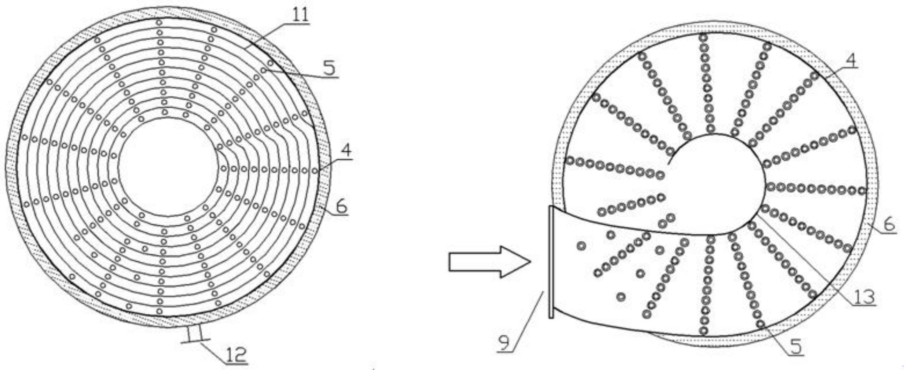


Figure 1: Sectional view of the new heat pipe heat exchanger



A-A cross-sectional B-B cross section

Figure 2: Section of the water side of the heat exchanger

l- flue gas outlet; 2- cold water inlet; 3- hot water outlet; 4- heat exchanger metal casing; 5- heat pipe with fins; 6-insulation layer; 7- metal base; 8- ash outlet; 9 -Flue gas inlet; 10-safety valve; l1-water side baffle; 12-sewage outlet; 13-water side baffle

4. CONCLUSIONS

Through the actual measurement of the kiln head exhaust gas of a new dry process cement production line of 3 × 2500t / d in a domestic cement plant, after understanding the production process of the cement plant, we focused on the analysis of the low temperature exhaust gas at the exit of the waste heat boiler of the current cement production line The potential for energy saving is as follows:

- The overall temperature level of the low-temperature exhaust gas at the outlet of the waste heat boiler is low, but the waste heat recovery potential is large.
- Heat exchange equipment suitable for low-temperature waste heat recovery needs to have some unique performance requirements.After investigating the current low-temperature waste heat recovery potential of the cement plant and the available heat exchange equipment status, this topic studied a new type of heat exchange equipment suitable for the low-temperature waste heat recovery of the cement plant. The new equipment can achieve both waste heat recovery and The purpose of dust removal.

NOMENCLATURE

A_0	area of the light pipe at 1m length	(m^2)
A_1	The external surface areas of the evaporation section	(m^2)
A_2	The external surface areas of the condensation section	(m^2)
A_f	Fin area on a 1m tube length	(m^2)
A_b	Bare tube area on a 1m tube length	(m^2)
A_w	windward area	(m^2)
A_{at}	actual heat transfer area	(m^2)
A_{re}	required circulation area of each tube	(m^2)
A'_{re}	area of each pipe	(m^2)
A_{aw}	Actual windward area	(m^2)
A_{01}	area of the first row of tubes	(m^2)
C_{p1}	Specific heat capacity	(kJ/ kg·°C)
d_v	Diameter determined by the speed of sound	(mm)
d'_v	Diameter determined by the speed of sound	(mm)
d_i	heat pipe diameter	(mm)
d_z	diameter of insulation	(mm)
d_0	outer diameter of the heat pipe	(mm)
f	coefficient of friction	(−)
G_{am}	Actual mass at the gas inlet	(kg/m^2·s)
G_2	inlet mass flow of water	(kg/m^2·s)
G_1	inlet mass flow of flue gas	(kg/m^2·s)
G_{a2}	actual mass flow rate of water	(kg/m^2·s)
G_{max}	mass flow rate of flue gas at the narrowest section	(kg/m^2·s)
g	gravitational acceleration	(m / s^2)
K_0	heat transfer coefficient	(W/m^2·°C)
K_1	Heat transfer coefficients of the evaporation section	(W/m^2·°C)
K_2	Heat transfer coefficients of the condensation section	(W/m^2·°C)

L	Total length	(m)
v_l	water flow velocity	(m/s)
Y	Fin gap	(mm)
α_1	heat dissipation coefficient	(−)
α_2	heat transfer coefficient	(−)
ρ_v	Density of steam	(kg/m^3)
ρ_l	density of liquid	(kg/m^3)
[L]	safe length ratio	(−)
L_j	economic length ratio	(−)
M_1	Flue gas mass flow	(kg / h)
M_2	Amount of hot water	(kg / h)
N	number of rows	(−)
n_p	Total number of pipes	(−)
P_v	Pressure of steam	(N / m^2)
ΔP	resistance of each row of tubes	(Pa)
q	heat transfer of each heat pipe	(kW)
Q_0	Total heat of the flue gas	(kW)
Q_1	Recovery of residual heat	(kW)
ΔQ	Heat loss	(kW)
Q_c	Heat transfer heat at the speed of sound limit	(kW)
Q_{ent}	limit heat transfer heat	(kW)
r	latent heat of vaporization	(kJ / kg)
S	pipe wall thickness	(mm)
S_1	tube spacing	(mm)
S_2	The length of the hypotenuse of the tube bundle	(mm)
T_1'	Flue gas inlet temperature	(℃)
T_1''	Flue gas outlet temperature	(℃)
T_2'	Water inlet temperature	(℃)
T_2''	Water outlet temperature	(℃)
T_{v1}	Temperature in the heat pipe at the flue gas inlet	(℃)
T_{v2}	Temperature in the heat pipe at the flue gas outlet	(℃)
T_1	hot fluid temperature	(℃)
T_2	cold fluid temperature	(℃)
$[T_v]$	working fluid saturation temperature	(℃)
t_0	ambient temperature	(℃)
σ	Surface tension	(N / m)
[σ]	allowable stress	(kgf/mm^2)
δ	Fin thickness	(mm)
β	wing ratio	(−)
λ_b	thermal conductivity of insulation material	(W/m·℃)

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