Journal of Thermal Science Vol.20, No.3 (2011) 248–253

DOI: 10.1007/s11630-011-0466-5 **Article ID: 1003-2169(2011)03-0248-06**

Analysis and Application of Variable Conductance Heat Pipe Air Preheater

Chengming Shi, Yang Wang, Quan Liao and Ying Yang

College of Power Engineering, Chongqing University, Chongqing, 400044, China

© Science Press and Institute of Engineering Thermophysics, CAS and Springer-Verlag Berlin Heidelberg 2010

The heat transfer analysis of variable conductance heat pipe air preheater was carried out. The temperature transfer matrix was obtained for the air preheater that comprises several discrete heat transfer units with same or different heat transfer surface area in a parallel or counter flow mode. By using the temperature transfer matrix, the outlet fluid temperatures could be easily calculated for a given air preheater and inlet fluid temperatures. The active length of condenser in a variable conductance heat pipe is determined according to the flat interface model. With the same initial conditions, the comparisons between variable conductance heat-pipe air preheater and regular heat pipe air preheater has been analyzed and tested in terms of heat pipe wall temperature, heat transfer surface area and outlet fluid temperatures. Based on the real industrial applications, it has been confirmed that the variable conductance heat pipe air preheater has excellent performance of anti-corrosion and anti-ash-deposition especially at the variable working condition and the sulfur coal (5%-6% mass fraction of sulfur) condition.

Keywords: variable conductance, heat pipe, air preheater, anti-corrosion, anti-ash deposition

Introduction

In the past several decades, most kinds of heat exchangers were theoretically or experimentally studied [1-3]. Since the heat transfer performance of all these heat exchangers is different, the expressions of heat transfer coefficient are different for the specific heat exchangers as well. Mostafa A et al[4] did research on heat transfer coefficient of heat pipe exchanger for heat recovery in an air conditioning. Due to the high heat transfer performance and adjustable wall temperature, the heat pipe exchangers were widely used as air preheater for a boiler, and a lot of researches[5-8] has been carried out in the past decades. Because of the special structure of heat pipe exchanger, the working fluids could exchange heat on the outer surface of heat pipe and the heat transfer surface of evaporation or condensation section could be absolutely separated. Since the wall temperature of heat pipe could be easily adjusted through the heat transfer surfaces of evaporation and condensation sections by choosing different fin coefficients, the acid corrosion of heat exchanger could be prevented or relatively reduced by adjusting the wall temperature of heat pipe above the dew point of flue gas. Therefore, the amount of ash deposition on the heat transfer surfaces could be reduced due to the higher wall temperature and the drier heat transfer surfaces. Moreover, since the fluids exchange the heat on the outer surface through heat pipe, the heat pipe exchanger is capable to prevent the leakage or mixing between fluids. Although the regular heat pipe exchanger could be excellently working at the design condition in which the wall temperature of heat pipe could be controlled by the ratio of heat transfer surface area between the evaporation and condensation sections of heat pipe and the acid corrosion or ash deposition on the surface of exchanger could be effectively prevented or relatively reduced, it can not satisfy the requirements of anti-corrosion and anti-ash-deposition when this regular heat

Received: December 2010
Chengming Shi: Associate Professor

pipe exchanger is working far from the design conditions, i.e., at the starting or shutting down process of a industrial boiler, at a variable working condition and with much higher mass fraction of sulfur coal than the designed coal. However, in a variable conductance heat pipe, not only the working fluid but also the inert gas was put into the inside volume of heat pipe. When a variable conductance heat pipe is working, the non-condensable inert gas was pushed at the end of condensation section by the working fluid steam. Since the volume of inert gas could be varied by the working pressure of heat pipe, a drop of evaporation temperature could result in a lower working pressure of heat pipe and it will lead the inert gas to expand; therefore, the effective condensation section of variable conductance heat pipe will be reduced and the wall temperature of heat pipe will be increased as compared to the regular heat pipe. Consequently, the low temperature corrosion of regular heat pipe could be prevented or relatively reduced[9]. Due to the self-adjusting wall temperature of variable conductance heat pipe upon the working conditions, the authors would like to take the advantage of regular heat pipe exchanger (such as excellent leak proof characteristics) and combine this unique characteristic of variable conductance heat pipe to improve the overall performance of exchanger in terms of anti-corrosion and anti-ash-deposition under low working temperature conditions.

The authors designed and applied this kind of variable conductance heat pipe exchanger with relatively small amount of inert gas for a 35t/h boiler of power plant in Chongqing, China. After 20 months' running, this variable conductance heat pipe exchanger obtained an excellent performance to prevent the ash deposition resulted from water condensation on the surface area; however, the surface area of last several rows of heat exchanger deposited a layer of rigid ash due to the acid corrosion, and the PH value of solution of ash and water with proportion of 1:6 is up to 1.0. Therefore, it could be concluded that the wall temperature of heat pipe must be higher than the flue gas acid dew point under all the working conditions in order to prevent acid corrosion and the condensation of water on the surface area of air preheater. Unfortunately, it is impossible for regular heat pipe exchanger to automatically adjust the wall temperature upon the variable working conditions to prevent the acid corrosion with the high sulfur coal, i.e., the mass fraction of sulfur is up to 5%~7%. However, it is believed that the variable conductance heat pipe exchanger with enough amount of inert gas will be an effective way to solve the problems such as acid corrosion and ash deposition on the rear rows of heat pipe. When a variable conductance heat pipe exchanger is working, the effective heat transfer surface areas of each heat pipe row are different due to the different operating temperature or pressure of each heat pipe row. Li et al[10] carried out the heat transfer calculation for a regular heat pipe exchanger at variable operating conditions with the same heat transfer surface area for each heat pipe rows. Up to now, there are rare reports about the heat transfer performance analysis of variable conductance heat pipe exchanger.

Therefore, the main task of this paper is to analyze the heat transfer performance of variable conductance heat pipe exchanger and find out whether the variable conductance heat pipe can effectively solve the anti-corrosion and anti-ash-deposition problems in the real industrial applications.

The active condensation length of variable conductance heat pipe

The active length of condensation

In literature [11], the heat transfer performance of variable conductance heat pipe i.e., VCHP (or called gas-filled heat pipe) was reported under an equilibrium condition, as shown in figure 1. The following assumptions are made for this variable conductance heat pipe: 1. The non-condensable inert gas complies the ideal gas law; 2. The heat pipe is operated under steady state conditions; 3. There exists a sharp interface between the non-condensable gas and working fluid. According to the ideal gas law, the filled inert gas submits to,

$$\frac{P_1 V_1}{T_1} = \frac{P_2 V_2}{T_2} \tag{1}$$

Where, P_1 is the pressure of filled gas at initial state, P_2 is working pressure of heat pipe, V_1 is the volume of inert gas at pressure P_1, V_2 is the volume of inert gas under pressure P_2, T_1 is the temperature of inert gas at P_1 and V_1, T_2 is the temperature of inert gas at P_2 and V_2.

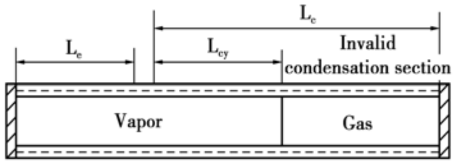

Fig.1 The schematic structure of variable conductance heat pipe at equilibrium condition

Since the inner diameter of heat pipe is usually a constant, under the working conditions, the length of heat pipe, i.e., L_q, occupied by inert gas could be evaluated as follows:

$$L_q = L\frac{P_1 T_2}{P_2 T_1} \tag{2}$$

Where, L is the length of heat pipe occupied by inert gas at the initial state. The value of L, T_1 and P_1 in the above equation could be considered as known parameters, and P_2 can be determined by iterative method when the active length of condensation is given. Therefore, L_q could be calculated from Eq. (2) when T_2 is known, and the active length of condensation of variable conductance heat pipe, L_{cy} could be expressed as,

$$L_{cy} = L_c - L_q \tag{3}$$

Where, L_{cy} is the active length of condensation for a variable conductance heat pipe; L_c is the total length of condensation section for a variable conductance heat pipe. Since the working pressure P_2, working temperature T_2 and active length of condensation for a variable conductance heat pipe are different for each of heat pipe rows, the values of P_2, T_2 and L_{cy} must be determined iteratively for each heat pipe row, respectively. Once the active length of condensation for a variable conductance heat pipe is known, the design calculation of a variable conductance heat pipe could be carried out by following the similar procedure for a regular heat pipe except by replacing L_c with L_{cy}.

Heat transfer analysis

In order to analyze the heat transfer performance of variable conductance heat pipe exchanger that comprises several independent variable conductance heat pipe rows, some simplifications and assumptions[12] about exchanger have to be made as follows,

1. the heat loss of heat pipe row is negligible;
2. the fins on the heat pipe continually across all the heat pipe rows;
3. the change of temperature is continuous along the flow direction;
4. all the individual heat pipe rows are assumed as an integrate heat transfer component which is same as a wall type heat exchanger.

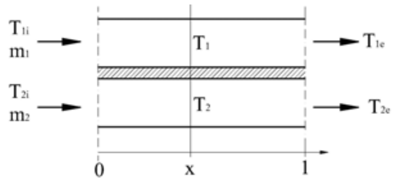

Fig.2 Heat transfer analysis under parallel flow arrangement

As shown in Fig.2, if the fluid flow along x direction, the quantity is positive value; otherwise, the quantity is minus.

$$\frac{dT_1}{dx} + K_1(T_1 - T_2) = 0 \tag{4a}$$

$$\frac{dT_2}{dx} + K_2(T_2 - T_1) = 0 \tag{4b}$$

Where:

$K_1 = 1/[(Re+Rc)m_1c_1]$; $K_2 = 1/[(Re+Rc)m_2c_2]$;

Re is the thermal resistance between hot fluid and working fluid in evaporation section, and Rc is thermal resistance between cold fluid and working fluid in condensation section.

Boundary conditions: $x=0$, $T_1=T_{1i}$, $T_2=T_{2i}$,
$x=1$, $T_1=T_{1e}$, $T_2=T_{2e}$,

If the heat loss between exchanger and ambient is negligible, the following equation could be obtained from the energy balance.

$$m_1c_1 dT_1 + m_2c_2 dT_2 = 0 \tag{5}$$

Assume M_1, M_2 are constant and integrate equation (5) in terms of x from 0 to 1.0, one obtains,

$$M_1(T_{1i} - T_1) + M_2(T_{2i} - T_2) = 0 \tag{6}$$

Where T_{1i}、T_{2i} are inlet temperatures of fluid 1 and 2, $m_1c_1=M_1$, $m_2c_2=M_2$,

Differentiate equation (4) and using equation (6), the second order linear, non-homogeneous differential equations with constant coefficients could be obtained as follows,

$$\frac{d^2T_1}{dx^2} + K_1\frac{dT_1}{dx} - Y_1T_1 + Z_1 = 0 \tag{7a}$$

$$\frac{d^2T_2}{dx^2} + K_2\frac{dT_2}{dx} - Y_2T_2 + Z_2 = 0 \tag{7b}$$

Where,

$$Y_1 = K_1K_2\frac{M_1+M_2}{M_2}, \quad Z_1 = K_1K_2T_{2i} + K_1K_2\frac{M_1}{M_2}T_{1i},$$

$$Y_2 = K_1K_2\frac{M_1+M_2}{M_1}, \quad Z_2 = K_1K_2T_{1i} + K_1K_2\frac{M_2}{M_1}T_{2i},$$

Solve differential equations (7), it yields,

$$T_1 = C_{11}\exp(a_1x) + C_{12}\exp(a_2x) + T_{01} \tag{8a}$$

$$T_2 = C_{21}\exp(a_3x) + C_{22}\exp(a_4x) + T_{02} \tag{8b}$$

Where,

$$a_1 = \frac{-K_1 + \sqrt{K_1^2 + 4K_1K_2\dfrac{M_1+M_2}{M_2}}}{2},$$

$$a_2 = \frac{-K_1 - \sqrt{K_1^2 + 4K_1K_2\dfrac{M_1+M_2}{M_2}}}{2},$$

$$a_3 = \frac{-K_2 + \sqrt{K_2^2 + 4K_1K_2\dfrac{M_1+M_2}{M_1}}}{2},$$

$$a_4 = \frac{-K_2 - \sqrt{K_2^2 + 4K_1K_2\dfrac{M_1+M_2}{M_1}}}{2},$$

$$T_{01} = T_{02} = Z_1/Y_1 = \frac{M_2}{M_1+M_2}T_{2i} + \frac{M_1}{M_1+M_2}T_{1i},$$

Apply the boundary conditions; the coefficients of equation (8) could be determined as follows,

$$C_{11}=\frac{T_{1\mathrm{i}}\exp(a_2)-T_{1\mathrm{e}}-T_{01}\exp(a_2)+T_{01}}{\exp(a_2)-\exp(a_1)} \tag{9a}$$

$$C_{12}=\frac{T_{1\mathrm{i}}\exp(a_1)-T_{1\mathrm{e}}-T_{01}\exp(a_1)+T_{01}}{\exp(a_1)-\exp(a_2)} \tag{9b}$$

$$C_{21}=\frac{T_{2\mathrm{i}}\exp(a_4)-T_{2\mathrm{e}}-T_{02}\exp(a_4)+T_{02}}{\exp(a_4)-\exp(a_3)} \tag{9c}$$

$$C_{22}=\frac{T_{2\mathrm{i}}\exp(a_3)-T_{2\mathrm{e}}-T_{02}\exp(a_3)+T_{02}}{\exp(a_3)-\exp(a_4)}, \tag{9d}$$

$T_{1\mathrm{e}}$、$T_{2\mathrm{e}}$ are outlet temperatures of fluid 1 and 2.

Since there is no special assumption for the exchanger, Equations (8) and (9) are the general analytical solution for the temperature distribution in a heat pipe exchanger.

Differentiate equation (8) and apply equation (4) and boundary conditions, one obtains,

$$C_{11}=\frac{K_1(T_{1\mathrm{e}}-T_{2\mathrm{e}})-K_1\exp(a_2)(T_{1\mathrm{i}}-T_{2\mathrm{i}})}{a_1[\exp(a_2)-\exp(a_1)]} \tag{10a}$$

$$C_{12}=\frac{K_1(T_{1\mathrm{e}}-T_{2\mathrm{e}})-K_1\exp(a_1)(T_{1\mathrm{i}}-T_{2\mathrm{i}})}{a_2[\exp(a_1)-\exp(a_2)]} \tag{10b}$$

$$C_{21}=\frac{K_2(T_{2\mathrm{e}}-T_{1\mathrm{e}})-K_2\exp(a_4)(T_{2\mathrm{i}}-T_{1\mathrm{i}})}{a_3[\exp(a_4)-\exp(a_3)]} \tag{10c}$$

$$C_{22}=\frac{K_2(T_{2\mathrm{e}}-T_{1\mathrm{e}})-K_2\exp(a_3)(T_{2\mathrm{i}}-T_{1\mathrm{i}})}{a_4[\exp(a_3)-\exp(a_4)]} \tag{10d}$$

Using equations (9) and (10), the temperature transfer matrix of heat exchanger could be obtained,

$$\mathrm{T_e}=\mathrm{AT_i} \tag{11}$$

Where, $\mathrm{T_e}=[\mathrm{T_{1e}},\mathrm{T_{2e}}]'$, $\mathrm{T_i}=[\mathrm{T_{1i}},\mathrm{T_{2i}}]'$

The elements of matrix A are,

$$a_{11}=\{K_1[(\exp(a_1)-\exp(a_2)]+a_2\exp(a_1)-a_1\exp(a_2)+[a_2-a_2\exp(a_1)+a_1\exp(a_2)-a_1]M_1/(M_1+M_2)\}/(a_2-a_1)$$

$$a_{12}=\{-K_1[(\exp(a_1)-\exp(a_2)]+[a_2-a_2\exp(a_1)+a_1\exp(a_2)-a_1]M_2/(M_1+M_2)\}/(a_2-a_1),$$

$$a_{21}=\{K_2[(\exp(a_4)-\exp(a_3)]+[a_4-a_4\exp(a_3)+a_3\exp(a_4)-a_3]M_1/(M_1+M_2)\}/(a_4-a_3),$$

$$a_{22}=\{K_2[(\exp(a_3)-\exp(a_4)]+a_4\exp(a_3)-a_3\exp(a_4)+[a_4-a_4\exp(a_3)+a_3\exp(a_4)-a_3]M_2/(M_1+M_2)\}/(a_4-a_3)$$

Where, the corresponding heat transfer surface area of elements in coefficient matrix is the whole surface area of exchanger.

Because the heat exchanger usually comprises multiple heat pipe rows and the fluid flow out from one heat pipe row and then enters into the near next heat pipe row, the outlet temperature of one heat pipe row is the inlet temperature of the next pipe row, the similar analysis method could be adopted for consecutive heat pipe rows with different surface area. Therefore, the temperature transfer matrix for each heat pipe rows could be obtained,

$$\mathrm{T_{1,e}}=\mathrm{A_1T_i},\ \mathrm{T_{2,e}}=\mathrm{A_2T_{1,e}},\ \ldots,\ \mathrm{T_e}=\mathrm{A}_n\mathrm{T}_{n\text{-}1,\mathrm{e}}$$

Based on the general analytical model for heat pipe exchanger, it is easy to get the temperature transfer matrix for the heat pipe exchanger with the same geometry of heat pipe rows:

$$\mathrm{T_e}=\mathrm{A_{row}}^n\mathrm{T_i} \tag{12}$$

Where, $\mathrm{A_{row}}=\mathrm{A_1}=\mathrm{A_2}=\ldots=\mathrm{A}_n$

Where, the matrix A_n is the temperature transfer matrix for n^{th} heat pipe row, and the corresponding heat transfer area of elements in above coefficient matrix is the area of each heat pipe row. Thus, if heat transfer area of each heat pipe row is different, the temperature transfer matrix is as follows,

$$\mathrm{T_e}=\mathrm{A_{ex}T_i} \tag{13}$$

Where, $\mathrm{A_{ex}}=\mathrm{A}_n\ldots\mathrm{A_3\,A_2\,A_1}$

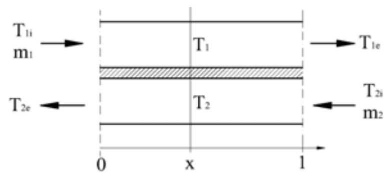

Fig.3 Heat transfer analysis under counter flow arrangement

As far as the heat pipe exchanger working in counter flow mode is concerned, the analytical model could be obtained by following the similar approach of parallel flow mode. As shown in Fig.3, the boundary conditions for counter flow arrangement are as follows,

x=0, $T_1=T_{1\mathrm{i}}$, $T_2=T_{2\mathrm{e}}$;

x=1, $T_1=T_{1\mathrm{e}}$, $T_2=T_{2\mathrm{i}}$.

In order to obtain the temperature transfer matrix for counter flow arrangement, one should carefully change the inlet temperatures in equation (11) and determine the value of fluid heat capacity according to the flow direction of this fluid as compared to the positive x direction, and then the temperature transfer matrix of counter flow arrangement could be written as follows,

$$\mathrm{T_e}=\mathrm{BT_i} \tag{14}$$

Where, $\mathrm{T_e}=[\mathrm{T_{1e}},\mathrm{T_{2i}}]'$, $\mathrm{T_i}=[\mathrm{T_{1i}},\mathrm{T_{2e}}]'$

Elements of the matrix B are as follows,

$$b_{11}=a_{11}-\frac{a_{12}a_{21}}{a_{22}},\ b_{12}=\frac{a_{12}}{a_{22}},\ b_{21}=-\frac{a_{21}}{a_{22}},\ b_{22}=\frac{1}{a_{22}}.$$

Where, the corresponding heat transfer surface area of element in above coefficient matrix is the surface area of each heat pipe row. If the individual heat pipe row of exchanger has the same heat transfer surface area, the temperature transfer matrix could be simplified as,

$$\mathrm{T_e}=\mathrm{B}^n\mathrm{T_i} \tag{15}$$

When the heat transfer surface area of each heat pipe row in an exchanger is different, the temperature transfer matrix can be written as,

$$T_e = B_{ex}T_i \tag{16}$$

Where, $B_{ex}=B_n\cdots B_2B_1T_i$

Equation (16) provides basis for designing calculation of variable conductance heat pipe exchanger working under counter flow condition. For a given exchanger, based on the inlet temperature of counter flow heat exchanger which is composed of several heat pipe rows with different heat transfer area, the outlet temperature can be easily calculated by equation (16).

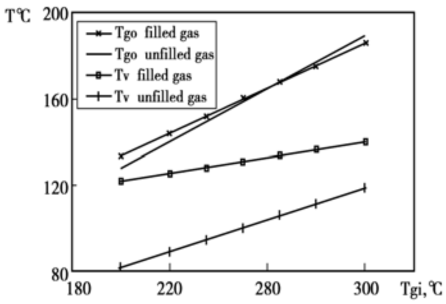

Fig. 4 The variation of the real outlet gas temperature and the lowest working temperature with inlet temperature

As shown in Fig. 4, it is presented that the comparisons of the lowest working temperature, T_v, and inlet flue gas temperature, T_{gi}, between regular heat pipe exchanger and variable conductance heat pipe exchanger. For these two kinds of exchangers, the designed flue gas temperature at outlet is 160℃; the lowest working temperatures of variable conductance heat pipe and regular heat pipe are 130℃ and 100℃, respectively. From this figure, it can be seen that the slope of outlet flue gas temperature, T_{go}, and the lowest working temperature of variable conductance heat pipe are smaller than those of regular heat pipe. Especially, as the increase of inlet flue gas temperature, the rate of change of working temperature for variable conductance heat pipe is much smaller than those of regular heat pipe, and this implies that the working temperature of variable conductance heat pipe will be in a relative small range under the relative wide range of working conditions for heat pipe exchanger. This unique characteristic of variable conductance heat pipe exchanger will be welcomed by the industrial boilers which are frequently operated at wide variable working conditions. For example, when the inlet flue gas temperature is 200℃, the working temperature of regular heat pipe is down to 82℃; however, the lowest working temperature of variable conductance heat pipe is still maintained above 122℃. Therefore, the variable conductance heat pipe has a unique self-adaptive capability at relatively low inlet flue gas temperature.

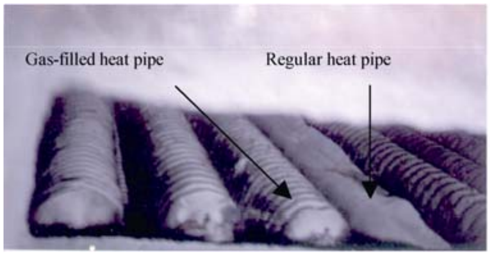

Fig. 5 The picture of heat pipe air preheaters of the three 20t/h boilers

According to the above mentioned design and analysis method for a variable conductance heat pipe exchanger, several variable conductance heat pipe air preheaters were designed and applied in the industrial boilers. As shown in Fig. 5, it is presented that the photo of heat pipe air preheater of three 20t/h boilers which were put into operation in Nov., 1996. The mass fraction of sulfur was 3% in the coal, and the designed outlet flue gas temperature was 160℃. In order to compare the performance of anti-corrosion and anti-ash-deposition between regular heat pipe and variable conductance heat pipe, one regular heat pipe was assembled with variable conductance heat pipe in this air preheater exchanger, as shown in Fig.5. At the first a couple of years operation, the load of boilers varied in a wide range (i.e., 4t/h~16t/h); therefore, the air preheater exchanger was working in a wide range of variable working conditions. After three years operation, the photo of outer surface of air preheater exchanger was taken in Nov. 1999, as shown in Fig.5, From this figure, it could be clearly seen that the regular heat pipe was fully deposited by the ash; however, the variable conductance heat pipes which are next close to the regular heat pipe were still relatively clean. Therefore, the advantages of variable conductance heat pipe about anti-corrosion and anti-ash-deposition are clearly confirmed as compared to the regular heat pipe.

Another example of comparisons about anti-corrosion and anti-ash-deposition between regular heat pipe and variable conductance heat pipe are shown in Fig.6. In this figure, it is presented the photo of heat pipe air preheat exchanger of two 35t/h boilers (BG-35/54-M_3) after one year operation. In this heat pipe exchanger, one regular heat pipe was assembled with other variable conductance heat pipe as well, and the flue gas flows from bottom to top in the air preheater. The mass fraction of sulfur in the coal is 3.34%, and the designed outlet flue gas temperature of the air preheater is 155℃. From the Fig.6, it could be clearly seen that the regular heat pipe which had the same geometry parameters with the variable conductance heat pipe almost became to a smooth pipe due to the acid corrosion ash deposition on the outer surface of heat pipe. Based on the above two industrial applications, the conclusions could be made that the variable conductance heat pipe undoubtedly has excellent performance on corrosion prevention and anti-ash deposition, especially in the case of cola which has high mass fraction of sulfur.

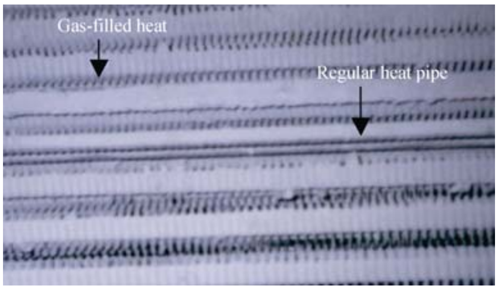

Fig. 6 The picture of heat pipe air preheaters of the two 35t/h boilers

Conclusions

Based on the above analysis and industrial applications, some conclusions could be made as follows,

1. The temperature transfer matrix of heat exchanger which is composed of several heat transfer units and working under parallel or counter flow mode is obtained. The relationship of inlet temperature and outlet temperature could be established through this temperature transfer matrix.

2. Since the variable conductance heat pipe exchanger has unique capability to self-adjust the working temperature, it is capable to be used as air preheater for industrial boilers with large variable operating conditions.

3. Based on the real industrial applications of variable conductance heat pipe exchanger for the high sulfur coal, it has been confirmed that the variable conductance heat pipe has undoubtedly excellent performance on anti-corrosion and anti-ash-deposition and these results could be used for engineering reference.

References

[1] JUNG-YANG SAN et a, Performance of a serpentine heat exchanger: Part I–Effectiveness and heat transfer characteristics[J], Applied Thermal Engineering, 2009, 29 (14-15): 3081–3087.

[2] DAVID KADYLAK et al. Effectiveness correlations for heat and mass transfer in membrane humidifiers[J], International Journal of Heat and Mass Transfer, Vol.52, No.5-6, 2009, pp1504–1509.

[3] B. Mathew, H. Hegab. Application of effectiveness-NTU relationship to parallel flow microchannel heat exchangers subjected to external heat transfer. International Journal of Thermal Sciences, Vol.49, No.1, 2009, pp1−10.

[4] Mostafa A. Abd El-Baky, Mousa M. Mohamed. Heat pipe heat exchanger for heat recovery in air conditioning, Applied Thermal Engineering, Vol.27, No.4, 2007, pp795–801.

[5] I.Kohtaka et al. Development, design and operation of large scale separate type heat pipe, Proc. of Sixth Int. Heat Pipe Conf, Grenoble, France, Vol.3, 1987, pp624.

[6] Jin Mingcong, Chen Yuanguo, Heat pipe and heat pipe exchanger[M], Chongqing University Press. Chongqing, China, 1986

[7] C.F. Than, K.S. Ong. Experiment and Prediction of Water –Filled Thermosyphon Heat Pipe Heat Exchanger Performance In Counterflow Air Streams Heat Pipe and Applications, Proceedings of the 13th International Heat Pipe Conference, Shanghai, China, 2005, pp568.

[8] Lukitobudi, A.R., Akbarzadeh, A., Johnson,P.W., & Hendy, P., Design, construction and testing of a thermosyphon heat exchanger formedium temperature heat recovery in bakeries,Heat Recovery Systems & CHP, Vol.15, No.5, 1995, pp481–491.

[9] Shi Chengming, Hu Huili et al, Designing analysis and application of air preheater with filled gas heat pipe, Journal of Engineering for Thermal Energy and Power, Vol.19, No.3, 2004, pp306–308

[10] Li Yourong et al, Calculation of heat pipe exchanger under variant working condition, Advance in application and research of heat pipe, The 3th national heat pipe conference symposium, Chongqing University Press, Chongqing, China, 1991, pp172.

[11] Marcus, B. D. Theory and design of variable conductance heat pipe, NASA CR, Report No.2018, 1972.

[12] Shi Chengming, Wang Yang et al, Performance analysis and application of three-fluid separate type heat pipe exchanger, Journal of Chongqing University, Vol.33, No.8, 2010, pp130–13.