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Research on a new type waste heat recovery gravity heat pipe exchanger

En Tian, Ya-Ling He, Wen-Quan Tao *

Key Laboratory of Thermo-Fluid Science and Engineering, Ministry of Education, School of Energy & Power Engineering, Xi'an Jiaotong University, Xi'an, Shaanxi 710049, People's Republic of China

HIGHLIGHTS

- Proposed a new type heat exchanger for waste heat recovery of gas.
- Completely solving the problem of gas-side easily to be blocked.
- Saving 15% natural gas without any blockage of the gas side channel.

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ABSTRACT

The industrial waste heat carrier, such as exhaust gas, mostly contains oil, particles, fibers and other impurities. If the conventional heat transfer enhancement techniques are applied in the gas side to recovery waste heat, the gas side flow channels are easily to be blocked, which not only greatly reduces the waste heat recovery efficiency but some time also makes the heat exchanger out of work. In this paper, a new type waste heat recovery heat pipe exchanger has been designed and applied to recover thermal energy in high temperature exhaust gas emitted from setting machine in the dyeing and printing industry. Its major feature is that clean air passes though fin-enhanced vertical tubes whose inner side is a condenser while dirty gas passes though inner smooth surface of horizontal tubes whose outside is an evaporator. The new type heat pipe exchanger has a big boiling chamber. The condensed water falls down to the chamber by gravity. Three-month continuous operation of recovering dirty exhaust gas waste heat shows that the new type heat pipe exchanger can save 15% natural gas without any blockage of the gas side channel.

1. Introduction

Energy is a very important resource in the social and economic development of the world. In the 21st century, the worldwide energy consumption has been rapidly increased, and the energy supply has become one of the important concerns of every government in the world. The large consumption of fossil fuels leads to severe environmental problems which is threatening the human life today. This is especially the case in China. In 2010, the energy consumption of China was about 3.27 billion tons of standard coal [1], having surpassed the consumption of U.S. and being the world's largest energy consumer. To make things worse, the energy utilization efficiency in China is quite low compared with the world average level. It is reported [2] that the energy efficiency of China is only about 33% at present, which is about 10% lower than that of the developed countries. As far as the energy consumption per unit GDP of China is concerned, it is two to three times higher than that of the world average, 2.4 times higher than that of United States, 4.4 times higher than that of Japan, 4.2 times higher than that of Germany [3]. Both the energy shortage and environmental pollution have highly required more effective use of energy. Therefore, the energy conservation and pollution reduction have been determined as the urgent basic state policy for economic and social development of China [4].

As it's well known, the low utilization rate of industrial waste heat is one of the important reasons for high energy consumption and low energy efficiency in China [5]. The amount of industrial waste heat is estimated to be 1.55 billion tons of standard coal in 2011, among which 60% could be recycled from the thermodynamics point of view, hence, as high as 930 million tons of standard coal could be saved every year. However, the waste heat recovery rate is only about 30% at present in China. If the waste heat recovery rate could be increased to 40%, the waste heat resource recycled can be increased by nearly 100 million tons of standard coal every year, which is very considerable. It should be noted that

* Corresponding author.
E-mail address: wqtao@mail.xjtu.edu.cn (W.-Q. Tao).

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Nomenclature

Latin symbols

D_{ev}	air side volume equivalent diameter, m
$d_{i,h}$	smooth tube internal diameter, m
$d_{o,h}$	smooth tube external diameter, m
d_o	base tube external diameter of finned tube, m
d_i	base tube internal diameter of finned tube, m
d_f	finned tube diameter, m
f_c	air side friction coefficient
G_{max}	air maximum mass flow rate, $kg/m^2 \cdot s$
g_c	gravity factor
h_h	exhaust gas side heat transfer coefficient, $W/(m^2 \cdot K)$
h_c	air side heat transfer coefficient, $W/(m^2 \cdot K)$
Q_h	exhaust gas side heat transfer rate, W
Q_c	air side heat transfer rate, W
l_f	fin height, m
l_h	smooth tube length, m
L_f	finned tube bundle length along flow direction, m
n_f	fin number per unit finned tube length
$S_{1,h}$	smooth tube bundle spanwise tube spacing, m
$S_{2,h}$	smooth tube bundle vertical tube spacing, m
$S_{1,c}$	finned tube bundle spanwise tube spacing, m
$S_{2,c}$	finned tube bundle vertical tube spacing, m
s_f	fins center distance, m
T_{h1}	exhaust gas inlet temperature, K
T_{h2}	exhaust gas outlet temperature, K
T_v	medium working temperature, K
T_{c1}	air inlet temperature, K
T_{c2}	air outlet temperature, K
T_{hf}	exhaust gas qualitative temperature, K
u_c	air inlet velocity, m/s
u_h	exhaust gas inlet velocity, m/s
V_h	exhaust gas flow rate, m^3/s
V_c	air flow rate, m^3/s

Greek alphabets

ΔP_c	air side pressure drop, Pa
λ_h	exhaust gas thermal conductivity, $W/(m \cdot K)$
λ_c	air thermal conductivity, $W/(m \cdot K)$
ΔT_h	exhaust gas side logarithmic mean temperature difference, K
ΔT_c	air side logarithmic mean temperature difference, K
δ_f	fin thickness, m
ρ_c	air density, kg/m^3
μ_c	air dynamic viscosity, $Pa \cdot s$
μ_w	air viscosity under heat pipe wall temperature, $Pa \cdot s$

Dimensionless groups

Pr_h	exhaust gas side Planck number
Re_h	exhaust gas side Reynolds number

the effective utilization of waste heat in industry is economically and environmentally very important not only for China but also for all countries over the world.

Many industrial-heating processes generate waste energy which is usually stored in hot gases with temperature around 150–300 degrees of Celsius. The emission of such hot gases not only wastes energy but also leads to thermal pollution to the environment. Hence the heat recovery of such hot gases is likely to be one of the major conservation methods to be adopted in a wide range of industries [6]. Both academic and industry communities have paid their attentions on this issue. A brief review for the papers published since 2000 is given below. An attempt was made by Yang using heat pipe exchanger for heating automobile using exhaust gas [7]. Liu et al. [8] investigated the performance of a looped two-phase separate heat pipe as waste heat recovery facility for the air-conditioning exhaust system. In Ref. [9], experimental study of heat pipe exchanger for cooling fresh air with returned air in air conditioning has been reported to investigate the thermal performance and effectiveness of heat recovery system. The performance of a waste heat recovery power generation system based on second law analysis was investigated in [10], which contributed further information on the role of gas composition, specific heat and pinch point influence on the performance of a waste heat recovery based on first and second law of thermodynamics [10]. The optimum finned pipe length at which maximum saving occurs for waste heat recovery systems was presented in [11]. A single-pass counter-flow heat exchanger was adopted to recover the shower water heat from bathrooms in residential buildings of Hong Kong for preheating cold water [12]. A new model was developed for determining the most appropriate waste heat recovery heat exchanger with maximum net gain in [13]. A new type of open-cell metal foam-filled plate heat exchanger based thermoelectric generator system (HE-TEG) was proposed to utilize low grade waste heat in [14]. The experimental study was conducted to investigate the heat and mass transfer characteristics in a titanium heat exchanger with excellent corrosion resistance used for waste heat recovery with the condensation arranged in a gas fired water heater [15], and the experimental results indicated that the thermal efficiency of the gas fired water heater with a latent heat recovery heat exchanger was enhanced by about 10% compared with heaters without heat recovery. An experimental study was conducted for the effects of increasing the back pressure on the engine performance by using two different types of heat exchangers, shell-and-tube-type heat exchanger and fin-and-tube-type heat exchanger, mounted in the exhaust line to recover waste heat from the engine exhaust [16]. It was found that the shell-and-tube-type exchanger could harvest approximately 1 kW more heat than that the fin-and-tube-type exchanger, and the back pressure generated by the shell-and-tube-type heat exchanger was approximately 2.5 kPa higher on average than that generated by the fin-and-tube-type exchanger. Another experiment was conducted using shell-and-tube heat exchanger to estimate the exhaust waste heat obtainable from a diesel engine in [17]. The exhaust waste heat recovery potential of a high-efficiency, low-emissions dual fuel low temperature combustion engine using an organic Rankine cycle (ORC) was examined in [18]. Stijepovic and Linke [19] have proposed a systematic approach to target waste heat recovery potentials and design optimal reuse options across plants in industrial zones, and the approach first established available waste heat qualities and reuse feasibilities considering distances between individual plants. Quoilin et al. [20] have developed a dynamic model of a small-scale organic Rankine cycle for recovering low-grade heat sources, and the results noted that small-scale ORCs were well adapted to waste heat recovery with variable heat source flow rate and temperature. Pandiyarajan et al. [21] experimentally studied the thermal energy storage from diesel engine exhaust gas using finned shell and tube heat exchanger. Mokkapati and Lin [22] conducted numerical investigation using corrugated tube heat exchanger with twisted tape inserts for exhaust heat recovery, and for a 120 ekW diesel generator. It was found that the application of corrugated tube with twisted tape concentric tube heat exchanger could save 2250 gals of fuel. Hatami et al. [23] investigated the effect of three designed heat exchangers (vortex generator heat exchanger, optimized finned tube heat exchanger and non-optimized heat exchanger) applied for diesel exhaust heat recovery. Clean and fouled aluminum and stainless steel exhaust gas recirculation-style heat exchangers was tested on a bench-scale thermoelectric apparatus to simulate automotive exhaust heat recovery by Love et al. [24]. It was discovered that heat exchangers fouled with diesel exhaust experience a degradation in performance of 5–10% compared with an unfouled heat exchanger of the same material. A heat pipe heat exchanger was used to recycle the waste heat in a slag cooling process of steel industry in [25]. The results indicated that the thermal performance parameters of the heat pipe heat exchanger decreased with running time of the system because that the calcium ions and sludge in waste water deposited on the outer surface of heat pipes, and after 14 days operation, the heat transfer rate, heat transfer coefficient, effectiveness and exergy efficiency varied from 7.27 to 6.19 kW; from 151.90 to 122.87 W/(m^2 K); from 0.14 to 0.12 and from 41.3% to 37.5%, respectively.

However, in all the above cited references for waste heat recovery of hot gases, the gas side heat transfer is deal with as heat transfer of clean medium. It is commonly known that the gas exhausted from many industrial systems generally contains some dirty materials, thus the most important character encountered in waste heat recovery is that the heat transfer media are dirty. However, this very important feature of waste heat recovery is often ignored in many publications. In a recent published monograph devoted for the application of low grade heat [26], all the intensification methods or techniques do not differ from the cases with clean working media. According to the authors' experiences, those techniques can hardly be used to dirty media encountered in practical engineering.

The goal of the present paper is to develop an effective way to recover the waste heat carried out by the hot gas in printing and dyeing industry, which is an important industrial section in the provinces along the seashore of China. In the printing and dyeing industry, the printing and dyeing processes for textile fabrics generally include singeing, desizing, scouring, bleaching and setting steps, in which the setting step is one of the most energy-intensive operations and also the main source of the air pollution. The setting machine is directly heated by the gases generated by burning natural gas for drying, neatening and finalizing the textile fabrics in the printing and dyeing industry. During the working process, a large amounts of high-temperature moist exhaust gas is emitted from setting machine which generally contains dust, oil mist, dyes auxiliaries and short threads of cotton and fibers. The exhaust gas temperature is generally around 445–455 K. Furthermore, the flow rate of the exhaust gas exhausted from one medium-size setting machine is about 15,000 m^3/h. That is to say, a large number of waste heat is directly emitted out from the setting machine, which causes not only the waste of energy but also the thermal air pollution. According to Ref. [27], the effective energy utilization rate of the setting machine is only 35% and the remainder is emitted into the atmosphere directly. Ogulata [28] also pointed out that the setting is the main step of the energy dissipation in textile industry and the waste heat recovery should be taken into consideration. Besides, Pulat et al. [6] investigated the potential of waste-heat obtained from dyeing process at textile industry in Bursa where textile center of Turkey and noted that there is a tremendous waste-heat potential to utilize in textile industry. In these three papers the potential of waste heat recovery for textile industry is pointed out, but no effective way of transfer heat to recover the waste heat was provided. If existing heat exchanger structure is used to recover the waste heat, the gas-side of the heat exchange is easily to be blocked after a short period of operation (example will be presented below).

Some efforts have been paid to improve the energy efficiency of this industry, and the gravity heat pipe exchanger is commonly adopted on the setting machine system (Fig. 1) for waste heat recovery [29], where the high temperature exhaust gas flows through the heat absorption end (i.e., evaporator section) of the gravity heat pipe exchanger. The working medium is vaporized at the heat absorption end and condensed at the condensing end, and finally flows back to the heat absorption end by its gravity for circulation. The clean air is sucked into the system, going through the condensing end and being heated. The schematic structure of one such the gravity heat pipe exchanger is shown in Fig. 2 [29].

There is another gas-liquid heat exchanger used for waste heat recovery designed based on the principle of heat pipe as shown in Fig. 3 [30]. The operating practice proved again that the heat exchanger would be easily blocked by dirty gas.

As mentioned above, the exhaust gas emitted from setting machine contains a mass of dust, oil mist, fiber and dye additives. Measurement has shown that for the case studied the exhaust gas contains 150–250 mg/m^3 particulate matter and 80–200 mg/m^3 lampblack. The combustion of natural gas produces large amount of water vapors and thus it increases the humidity of exhaust gas significantly. The high humidity exhaust gas with dust, fiber is very easy to adhere to the fin and tube wall when it is flowing within finned tube banks. Even for smooth tube bank such dirty flow media will quite soon blocked the space between tubes, not only significantly reducing the waste heat recovery efficiency [27], but even also stopping the system operation. Fig. 4 presents a picture displaying how the hot gas channel of the waste heat recovery heat pipe exchanger is blocked by fiber oil. This is a typical example showing that those effective enhanced techniques for clean media cannot be used for dirty heat transfer media encountered in the waste heat recovery.

Stimulated by the above difficulties, through several years engineering practice we figured out a practical configuration of heat exchanger which can be effectively used to recover the waste heat contained in dirty gas. Its major feature is that clean air passes through fin-enhanced vertical tubes whose inner side is a condenser while dirty gas passes through inner smooth surface of horizontal tubes whose outside is an evaporator. The new type heat exchanger has a big boiling chamber. The condensed water falls down to the chamber by gravity. Because the exhaust gas with enough large velocity is difficult to stick and block as flowing through the inner surface of smooth tubes, the new type heat pipe exchanger can effectively solve easy-blockage problem. In addition, the new type heat pipe exchanger can operate stably for a relatively long-term. In the following we will present its structure, thermal design and some results of application.

2. New type waste heat recovery heat pipe exchanger

The gravity heat pipe has been widely used in many industry fields because of its simple structure and economical cost [31–34]. When it is going to be applied for the waste heat recovery, one thing must be always kept in mind that at the dirty media side the application of conventional enhanced structure should be very careful.

In this paper, a new type heat pipe exchanger has been proposed for exhaust gas waste heat recovery on setting machine by employing the working principle of gravity heat pipe. Its pictorial picture is shown in Fig. 5. The whole heat exchanger is a big connected chamber with smooth tube bundle acting as the heat absorption (evaporating) end while finned tube bundle acting as

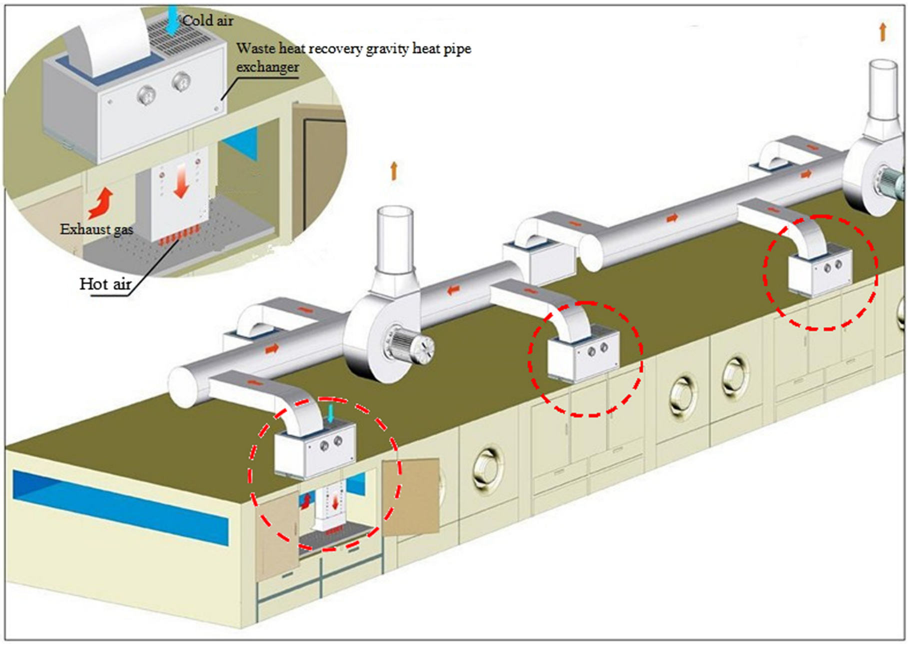

Fig. 1. The original gravity heat pipe exchanger waste heat recovery system.

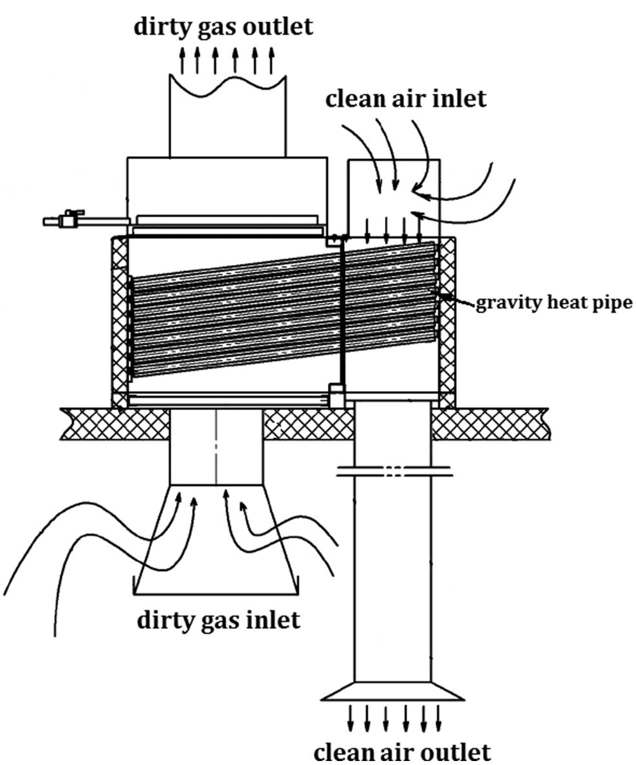

Fig. 2. The schematic of exhaust gas gravity heat pipe exchanger in setting machine.

the heat release (condensing) end. This heat exchanger is actually composed of two parts: the lower part is a kind of shell-and-tube heat exchanger, while the upper part is a duct for air flow with a bundle of finned tubes. To enhance the air side heat transfer the spirally finned tubes are adopted. There is a separating wall between the two parts. The heat exchanger working process is as follows. First the high temperature exhaust gas flows through the inner side of the smooth tubes and transfers heat to outside working medium (water). In the shell side a certain amount of water is kept and heated by the tubes immersed within it. The water vapor rises and is further heated by the rest of tubes within the shell. Then the superheated vapor goes into the inner side of the finned tubes which are connected with the separating wall. The vapor condenses by releasing latent heat of vaporization to the outside clean air through the finned tube bundle wall, and then falls down to the bottom of the heat exchanger by gravity. Thus the clean air is pre-heated by the exhaust gas, and is sucked into the setting machine drying oven for combustion.

From the above description, it can be seen that the novelties of this new type heat pipe heat exchangers are of threefold. First, it is not a conventional heat pipe, rather it is a separated heat pipe, which means that the evaporator and condenser are not in the same tube. Second, it is not a traditional separated heat pipe, for which the condensation and boiling are both occur in tubes [35]. Here boiling is occur in a big pool which is in favor of stabilizing the boiling heat transfer compared with the boiling within tube. Third, gas with dirty materials is flowing inside smooth tube with enough velocity, whose scrubbing effect helps the gas flow carrying the dirty materials out of the tube. And smooth tube surface is easy to be cleaned compared with finned surfaces.

3. Thermal design process

3.1. Known parameters and design steps

The new type heat pipe exchanger is made of steel with the tube bundles in staggered array. The given condition for the thermal design is presented in Table 1. As shown in the table, the hot gas temperature of 448 K is required to be cooled with a flow rate of 0.5 m^3/s. The inlet temperature of clean air is 308 K and its flow rate is 0.2 m^3/s. The clean air side will adopt finned tube for enhancing heat transfer. The allowed air side pressure drop is 20 Pa. According to the manufacturing condition for the practical case studied, the tube side fin dimensions are selected as shown in the table (also see Fig. 6).

The main task of designing the new type heat pipe exchanger is to determine the clean air side heat transfer area and tube numbers

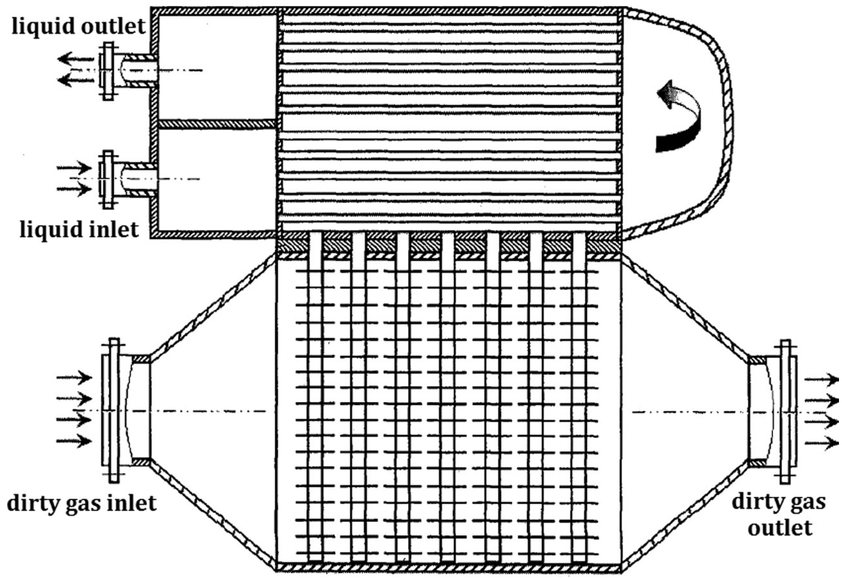

Fig. 3. Gas-liquid heat exchanger for recovering waste heat.

Fig. 4. Waste heat recovery heat pipe exchanger blocked by fiber oil.

for the evaporation part. The given conditions are provided from the engineering system which has a unique feature different from conventional heat exchanger design described in textbooks [31–34], in that neither the heat transfer area nor the heat transfer rate is specified. Meanwhile the gas side pressure drop is specified which gives a constraint for the gas side flow rate and heat transfer area. After some preliminary try-and-error calculation we finally figure out following thermal design steps:

(1) Assume the exhaust gas outlet temperature, and determine the physical parameters of exhaust gas by the exhaust gas reference temperature (i.e., arithmetic mean temperature), and then calculate the exhaust gas side heat transfer rate according to its enthalpy difference between inlet and outlet of the heat exchanger. The gas thermal physical properties are taken from Ref. [31].
(2) Assume the air physical parameters and calculate the air outlet temperature according to the gas side heat transfer rate. Then with the calculated air outlet temperature, proofread the air physical parameters, and then revise the air outlet temperature with updated air physical parameters. Repeat above calculation procedure until the difference between two successive calculations is less than 1%.
(3) Select the saturation temperature of the working medium and determine the logarithmic mean temperature difference for both the exhaust gas side and the air side.
(4) Calculate the exhaust gas side and the air side heat transfer coefficient according to the experimental correlations, respectively.
(5) Assume that both condensation and boiling heat transfer coefficient of the water side are large enough that their thermal resistances can be neglected. Determine the required heat transfer area for both the exhaust gas side and the air side based on the gas side and air side heat transfer coefficients, respectively.
(6) Determine the total numbers of the smooth tube and the finned tube.
(7) Check the gas side pressure drop, which should be less than the allowable value.
(8) If the calculated gas side pressure drop is too large, depending on how large the difference, re-design can be conducted along following lines:
(a) If the difference is less than 20%, reduce the gas side tube length (that means reduce the tube length along gas flow direction) by approximately 20% and redo the whole calculation.
(b) If the difference is larger 20%, apart from reducing the gas side tube length the in-tube average gas velocity should be reduced by a certain amount. However in order to keep a strong scrubbing effect of the in-tube gas flow on the carried dirty material this average gas velocity should not be less than 8 m/s.

It should be noted that the satisfaction of gas-side pressure drop is the major constraint of this design. The revision of gas-side structure may cause some modification of air-side heat transfer area. The details are omitted here for simplicity. The experiment study of this paper is for verifying the idea of the new type waste heat recovery heat pipe exchanger. There are some rooms for optimization of the gas-side and air-side geometries and selection of suitable phase change heat transfer medium. These will be conducted in our next stage of investigation.

3.2. The thermal design calculation process

In this design calculation, the following experimental correlations are used:

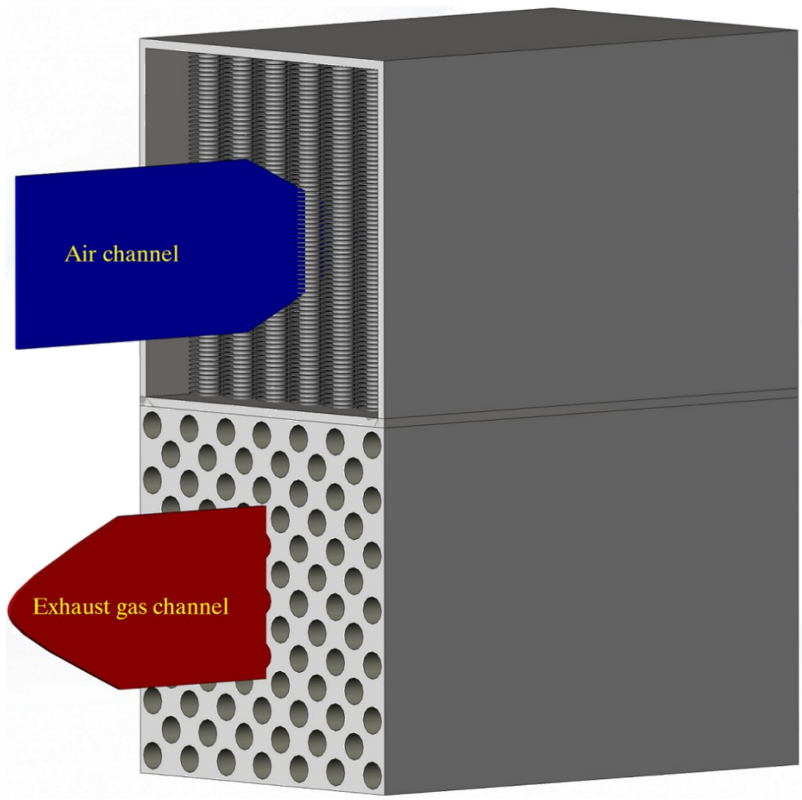

Fig. 5. The Schematic diagram of new type waste heat recovery heat pipe exchanger.

Table 1
Known parameters.

Δp_c	⩽20 Pa	V_h	0.5 m^3/s	δ_f	0.001 m
$d_{i,h}$	0.029 m	T_{c1}	308 K	s_f	0.005 m
$d_{o,h}$	0.032 m	V_c	0.2 m^3/s	n_f	200 m^{-1}
$S_{1,h}$	0.048 m	d_o	0.025 m	d_f	0.045 m
$S_{2,h}$	0.032 m	d_i	0.02 m	$S_{1,c}$	0.07 m
T_{h1}	448 K	l_f	0.01 m	$S_{2,c}$	0.08 m

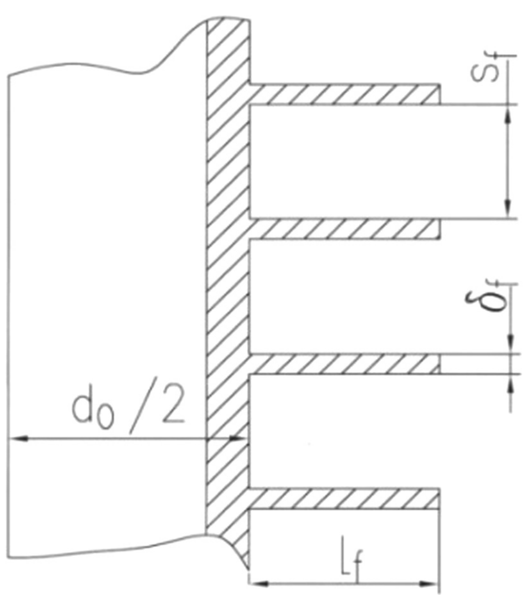

Fig. 6. Heat pipe finned by spiral-fins.

The average heat transfer coefficient for the in-tube gas turbulent flow is determined by Gnielinski formula [31]:

$$h_h = (\lambda_h/d_{i,h})(f_h/8)(Re_h - 1000)Pr_h/(1 + 12.7\sqrt{f_h/8}((Pr_h)^{2/3} - 1)) \times (1 + (d_{i,h}/l_h)^{2/3})(T_{hf}/T_v)^{0.45}$$

$$f_h = (1.82\lg Re_h - 1.64)^{-2} \quad Re_h = 2300 - 10^6, Pr_h = 0.6 - 10^5 \tag{1}$$

The Gnielinski formula is the most accurate one for calculating heat transfer coefficient of the in-tube gas turbulent flow, and this fact has been well-recognized by recently published heat transfer textbooks (see references [31–34]). In the basis of more than 800 experimental data, the deviations of the most data are within 10%.

The average heat transfer coefficient for air flow across finned tube bundle is calculated by following equation [36]:

$$h_c = 0.27((S_{1,c}/d_o - 1)/(((0.5S_{1,c})^2 + (S_{2,c})^2)^{0.5}/d_o - 1))^{0.2}\lambda_c/s_f(d_o/s_f)^{-0.54} (l_f/s_f)^{-0.14}(u_c s_f/v_c)^{0.65} \quad S_{2,c}/d_o > 2 \tag{2}$$

Fig. 7. Picture of new type heat pipe exchanger.

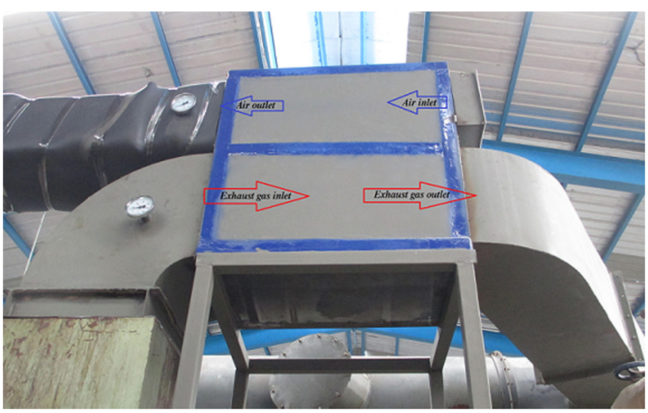

Fig. 8. New type heat pipe exchanger test system.

It is interested to note that Ref. [36] is the translation of design standard of boiler unit issued by former Soviet Union whose reliability and feasibility has been widely validated.

The pressure drop through the tube bundle finned by spiral-fins is calculated by Gunter and Shaw formula [37]:

$$\Delta P = \frac{f_c G_{max}^2 L_f}{2 g_c D_{eV} \rho_c} (\mu_c/\mu_w)^{-0.14} (D_{eV}/S_{2,c})^{0.4} (S_{1,c}/S_{2,c})^{0.6} \tag{3}$$

Although this formula was published years ago, its applicability is well recognized, see [38] for example.

Some important results of the thermal design are presented as follows: the number of finned tube is 40 with length 0.35 m and the number of smooth tube is 90 with length 0.74 m. In addition, the overall length, width and height of the new type heat pipe exchanger are 0.74 m, 0.4 m and 0.75 m, respectively. The average gas velocity in tubes is 8.2 m/s. The predicted outlet gas and air temperatures are 422 K and 365 K, respectively. Besides, the calculated air side pressure drop is 18.14 Pa. For saving space, the calculation details are not included in this paper.

4. Experiment process

4.1. Test results

The new type heat pipe exchanger is manufactured according to the thermal design calculation and its outline is shown in Fig. 7. Five new type heat pipe exchangers are installed on one setting machine of a printing and dyeing factory in Shaoxing city, Zhejiang Province, China for testing heat exchanger performance (Fig. 8). A kind of cloth called Gaonianlanma is taken to be set for which during the setting process fibers with different lengths can be easily produced. The measured process parameters and the correspondent methods are as follows. The heat exchanger inlet and outlet temperatures of clean air and the exhaust gas are measured by portable metal thermometers with maximum temperature of 300 °C are adopted. The Pitot tube is used to measure the air and exhaust gas inlet velocity. Since the original heat exchanger could not last long before the gas side channels were fully blocked, no comparison test could be arranged between the original and the new one. In order to reveal the energy-saving function of the new heat exchanger two series tests were conducted. In the first series the five new heat exchanger were installed and the gas fuel consumption was carefully recorded. In the second series of test the new type heat exchanger were totally switched off the system but the same production quantity of cloth. The operating parameters of the five new type heat exchangers and the natural gas consumption with and without using new type heat pipe exchanger are presented in Tables 2 and 3, respectively. During the experiments the setting machine speed and drying oven temperature were maintained nearly unchanged.

It can be seen from Table 2 that the deviation of the measurement result of the average exhaust gas outlet temperature from the design value is only 1.43%, indicating very good agreement. Moreover, it can also be easily observed from Table 3 that the five new type waste heat recovery heat pipe exchangers can save about 15% natural gas for one setting machine.

4.2. Check of anti-blocking effect of new type heat exchanger

After three-months continuous and stable operation, the exhaust gas side of new type heat pipe exchanger was switched off the system and its gas side surface was checked. As shown in Fig. 9, there is no any sign of adhesion or congestion of fibers on

Table 2
Operating parameters of new type heat pipe exchanger.

New type heat exchanger No.	1	2	3	4	5
Exhaust gas inlet temperature (K)	465	459	448	446	448
Exhaust gas outlet temperature (K)	441	435	424	421	423
Air inlet temperature (K)	308	308	308	308	308
Air outlet temperature (K)	375	365	368	361	364
Exhaust gas flow velocity (u_h, m/s)	8.0	7.8	7.4	7.5	8.2
Air inlet velocity (u_c, m/s)	1.4	1.8	1.6	2.0	1.7
Temperature deviation between measurement result and design value (%)	1.08	1.57	2.65	1.72	0.13

Table 3
Natural gas consumption with and without new type heat pipe exchanger (3 h continuous measurement).

Heat exchanger air inlet conditions	Natural gas average instantaneous dosage (m^3/h)	Natural gas accumulated dosage (m^3/h)
Without	86.5	278.3
With	73.4	229.7

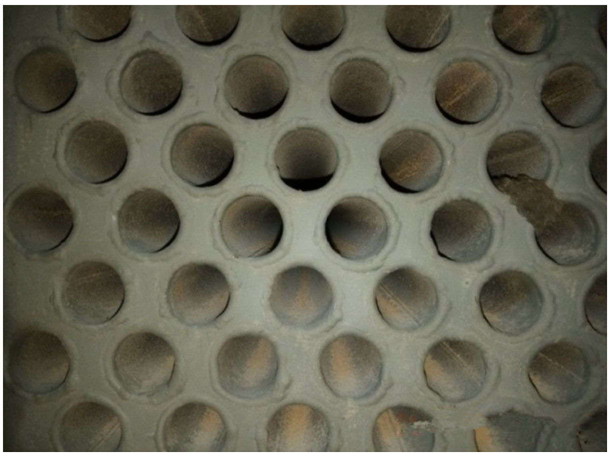

Fig. 9. Exhaust gas side condition of new type heat pipe exchanger after three-month continuous operation.

the tube sheet and tube inner surfaces. Even though some tiny particle sediment can be found on the surface which does affect the heat transfer, but this is much better than channels being blocked. In addition after several months operation the heat exchangers may be switched off the system and with an aid of some metal brush such sediment is easily to be scraped off the surface.

5. Discussion

There are more than 2000 setting machine systems and more than 1.65 million cubic annual consumption of natural gas for each setting machine in Shaoxing city [27]. After the application of the new type waste heat recovery heat pipe exchanger for all the setting machine system, approximately 600,000 tons of standard coal could be saved at least each year for Shaoxing city.

Dirty with some adhesive materials is the basic feature of most exhaust gas which often contains an appreciable amount of waste heat. How to effectively recover the waste heat from such dirty media is a big challenge not only for China but also for other counties in the world as well. Present paper proposed one effective way and further studies are highly required in this aspect.

As indicated above, in the second phase of our study some further works will be conducted, including improvement of structure. The present new type heat pipe exchanger is a big vacuum boiling chamber, and its manufacture needs high welding technique. One of our future work will focus on the modular design, i.e., the whole heat exchanger is divided into two-four subsets, such that if one of the subset fails to work replacement is only needed for this subset.

6. Conclusions

In this paper, a new type heat pipe air-gas heat exchanger has been proposed, designed and applied successfully. Its major feature is that clean air passes though fin-enhanced vertical tubes whose inner side is a condenser while dirty gas passes though inner smooth surface of horizontal tubes whose outside is an evaporator. Three-months continuous operation of recovering dirty exhaust gas waste heat shows that the new type heat pipe exchanger can save 15% natural gas without any blockage of the gas side channel, thus completely solving the problem of the original heat exchanger for which gas-side space was easy to be blocked within several hours operation because of the finned surface adopted for the dirty gas side.

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