Closed-ended oscillating heat-pipe (CEOHP) air-preheater for energy thrift in a dryer

S. Rittidech *, W. Dangeton, S. Soponronnarit

Faculty of Engineering, Mahasarakham University, Thailand

Abstract

The CEOHP air-preheater consisted of two main parts, i.e. the rectangular house casing and the CEOHP. The house casing was designed to be suitable for the CEOHP. The inside house casing divided the CEOHP into three parts, i.e. the evaporator, the adiabatic section and condenser section. The CEOHP air-preheater design employed copper tubes: thirty-two sets of capillary tubes with an inner diameter of 0.002 m, an evaporator and a condenser length of 0.19 m, and each of which has eight meandering turns. The evaporator section was heated by hot-gas, while the condenser section was cooled by fresh air. In the experiment, the hot-gas temperature was 60, 70 or 80 °C with the hot-gas velocity of 3.3 m/s. The fresh-air temperature was 30 °C. Water and R123 was used as the working fluid with a filling ratio of 50%. It was found that, as the hot-gas temperature increases from 60 to 80 °C, the thermal effectiveness slightly increases. If the working fluid changes from water to R123, the thermal effectiveness slightly increases. The designed CEOHP air-preheater achieves energy thrift.

Keywords: Closed-ended oscillating heat-pipe air-preheater; Energy thrift dryer

* Corresponding author. Tel.: +66-43-754316; fax: +66-43-754316.
E-mail address: s_rittidej@hotmail.com (S. Rittidech).
1. Introduction

1.1. Target

The effort to reduce energy consumption is a main goal for the dryer. Traditionally the dryer is unable to use its waste heat. Compared with other conventional heat-exchangers, the closed-ended oscillating heat-pipe (CEOHP) heat-exchanger has many advantages, e.g. large quantities of heat are transported through a small cross-section area. The CEOHP is a very effective heat-transfer device [1]: it has a simple structure and a fast thermal-response. The CEOHP consists of a long capil-
lary tube bent into many turns and the evaporator, adiabatic section and condenser section are located at these turns. However, there is no wick structure to return the condensate liquid from the condenser to the evaporator section. Heat is transported from the evaporator section to the condenser section by the pulsation of the working fluid moving in an axial direction in the tube. The inner diameter of the pipe is important. It must be small enough so that, under operational conditions, liquid slugs and vapor plugs can be formed. If the diameter is too large, the liquid and vapor inside the tube will become stratified and operation cannot be established. Rittidech et al. [3] investigated the effect of inclination angles, evaporator lengths and working-fluid properties on the heat-transfer characteristics of the CEOHP under normal operating condition. Rittidech et al. [4] devised a correlation to predict the heat-transfer characteristics of a CEOHP. Yang et al. [5] studied a water heat-pipe heat-exchanger using automotive-exhaust gas. The experimental results indicate it is worthwhile using a heat-pipe heat-exchanger employing the exhaust gas.

The application of CEOHPs, in order to save energy, is considered. The CEOHP can have many applications such as, economizers or air preheaters, electronic-cooling devices and solar collectors to name just a few. The new type of heat-pipe or CEOHP, shown in Fig. 1, can help in energy saving, specifically for air preheating in a dryer. The objective of the present study is to design, construct and test the CEOHP air-preheater for a dryer that can recover the waste heat from the drying process.

1.2. Conventional-drying process (see Fig. 2)

First, the air is heated by fuel combustion. Then the hot air moves through the layers of the product. The heat is transferred to the product and evaporation occurs.
2. CEOHP air-preheater system – design concept and calculation

Some fundamental design parameters e.g. the maximum heat-transfer rate of the drying system ($Q_{\text{max}}$, $W$) and the installation site depending on the available space have to be considered. Hence for the counterflow air-preheater heat-exchanger, the following set of equations applies [6]:

$$Q_{\text{max}} = C_{\text{min}}(T_{\text{hi}} - T_{\text{ci}}),$$
$$C_{\text{min}} = \rho V A C_p.$$  \hspace{1cm} (1)

2.1. CEOHP air-preheater design

When considering the design of the CEOHP air-preheater, the size and position of the air-preheater are taken into consideration, as well as the maintenance and budget costs. In addition, insulating the ducts and air-preheater also decreases the heat loss. The CEOHP air-preheater heat-transfer design was solved in six steps as follows:

- The hot-gas temperature of initial and fresh air temperature from drying process is obtained.
- The working temperature is calculated and the tube material is selected.
- The type of working fluid that is appropriate to the operational temperature.
- The inner diameter of the CEOHP is calculated from Maezawa et al. [2]:
  $$D_{\text{max}} \leq \sqrt{\frac{\sigma}{\rho g}},$$  \hspace{1cm} (2)
- $L_e$, $L_n$, $L_c$, $L_t$, and $n$ are defined corresponding to the duct size of the dryer and the installation area.
• $Pr_v$, $\rho_v$, $\rho_l$, $h_{fg}$ and the oscillation phenomena of the working fluid at working temperature are described and the heat flux of CEOHP air-preheater is solved by the correlation from [4]. The standard deviation of this equation is ±30%.

\[
Ku_{90} = 0.0067 \left[ \left( \frac{D_{1.1}^{1.1} L_{t_1}^{0.1}}{L_{c}^{3.2}} \right) n^{0.9} Pr_{v}^{12} \left( \frac{\rho_v}{\rho_l} \right)^{-0.1} \left( \frac{\omega \mu_v^3}{\sigma^2 \rho_v} \right)^{0.01} \right]^{0.175}.
\]  

(3)

It was necessary to change from the heat flux ($q$) from to the heat-transfer ($Q$)

\[
Q_{90} = (Aq_{90}).
\]  

(4)

From this correlation, the meaning of each parameter can be discussed as follows:

$Ku_{90}$ indicates the ratio of heat flux to critical heat flux for the vertical orientation. $D_{1.1}^{1.1} L_{t_1}^{0.1} / L_{c}^{3.2}$ defines the size of the CEOHP. For example, if the value of $D_{1.1}^{1.1} L_{t_1}^{0.1} / L_{c}^{3.2}$ was very high, then the tube would be large and the evaporator section would be short. Because of the boiling phenomenon, the value of $Ku_{90}$ or heat flux would be high. If the value of $D_{1.1}^{1.1} L_{t_1}^{0.1} / L_{c}^{3.2}$ was very low, then the tube would be small and the evaporator section would be long. Because the boiling phenomenon within this type of tube will be akin to the boiling phenomenon in a confined channel, the values of $Ku_{90}$ or heat flux will be low.

The number $n$ indicates that of the capillary tubes which connect the evaporator and the condenser section of a CEOHP or represents the number of turns of a CEOHP; $Pr_v$ indicates the ratio of momentum diffusivity to the thermal diffusivity of vapor slug. If the value is very low, the vapor slug will be able to transfer the thermal energy to the condenser section relatively efficient. Therefore, the value of $Ku_{90}$ or heat flux will be high.

The ratio $\rho_v/\rho_l$ indicates the vapor phase density to liquid phase density of the working fluid that represents the working pressure the working fluid within the CEOHP.

The oscillation phenomena within the CEOHP, can be described by $\omega \mu_v^3 / \sigma^2 \rho_v$, i.e.

\[
\frac{\omega \mu_v^3}{\sigma^2 \rho_v} = \left( \frac{g \mu_v^4 / \rho_v \sigma^3}{\rho_v L_v \sigma / \mu_v^2} \right)^{0.5} \left( \frac{\rho_l}{\rho_v} \right)^{0.5},
\]

where the capillary buoyancy number is $(g \mu_v^4 / \rho_v \sigma^3)$ and the Suratman number $(\rho_v L_v \sigma / \mu_v^2)$.

The capillary buoyancy and Suratman numbers are the ratios of surface tension force and the viscous force. The frequency $\omega$ of the oscillating motion of the vapor slug in a tube is defined as the frequency of simple harmonic motion, i.e. $\omega = \sqrt{\rho_g / \rho_v L_v}$.

Values of some of these parameters are shown in Table 1.

The effectiveness of CEOHP air-preheater, $\varepsilon$, is defined as the ratio of the actual heat-transfer rate for an air-preheater heat-exchanger to the maximum possible heat-transfer rate, i.e.,
\[ e = \frac{Q_{\text{Actual}}}{Q_{\text{Max}}}, \]  

(5)

where

\[ Q_{\text{Actual}} = \rho V A C_p (T_{co} - T_{ci}). \]  

(6)

The results of the calculation are shown in Table 2.

### 3. Experimental set-up

#### 3.1. Prototype

The CEOHP air-preheater for the drying process, see Fig. 3, was divided into two parts, i.e. the evaporator and condenser section with the length of 0.19 m. The tube arrangement was aligned in the direction of the hot-gas flow. The condenser section was connected to the fresh-air section and the evaporator section was in contact with the heat source from the gas burner. The dryer-bath type was selected to be appropriate to the CEOHP air-preheater. The prototype has a duct size area of 200 mm × 200 mm including the fibre insulation.

#### 3.2. Test rig

This prototype was installed in a test rig, as shown in Fig. 4. The hot-gas coming from the gas burner flows through the CEOHP air-preheater. The initial and final temperatures are measured with thermocouples. Twelve thermocouples type K were
installed on the evaporator section and twelve more on the condenser. These thermocouples were connected to a Yokogawa-MX100 acquisition data-system. When a steady state was achieved, the temperatures at the inlet and outlet of the evaporator and the condenser section were recorded. The heat-transfer rate and effectiveness were determined and compared with the predicted values.

The controlled parameter was the hot-gas velocity of 3.3 m/s
The variable parameters were:

- working-fluid water or R123,
- hot-gas temperature of 60, 70 or 80 °C.

Fig. 3. The prototype: (a) CEOHP air-preheater; (b) batch-type dryer; (c) dryer with CEOHP air-preheater.

installed on the evaporator section and twelve more on the condenser. These thermocouples were connected to a Yokogawa-MX100 acquisition data-system. When a steady state was achieved, the temperatures at the inlet and outlet of the evaporator and the condenser section were recorded. The heat-transfer rate and effectiveness were determined and compared with the predicted values.

The controlled parameter was the hot-gas velocity of 3.3 m/s
The variable parameters were:

- working-fluid water or R123,
- hot-gas temperature of 60, 70 or 80 °C.
4. Results and discussion

Table 3 shows the results of the experiment.

4.1. Effect of hot-gas temperature on heat-transfer rate

In this experiment, the CEOHP air-preheater had eight turns, 32 columns, and a hot-gas velocity of 3.3 m/s. The experimental results present the effect of the hot-gas temperature on the heat-transfer rate – see Fig. 5. This figure compares the experimental results with the predictions from the correlation [3]. It can be seen that, when the hot-gas inlet temperature increases, the heat-transfer rate also rises. This is because when the hot-gas inlet-temperature increases, the fresh-air outlet-temperature also increases. Thus, the temperature difference between the inlet and outlet air temperature also increases and the actual heat-transfer rate will be high. The heat-transfer rate as measured was lower than that predicted via the correlation [3]. However,
the predictions compare well with experimental data for the 80 °C run. In addition, when the hot-gas temperature increased from 70 to 80 °C the experimental data were within the standard deviation of ±30% from the correlation predictions. It can be concluded that, if the hot-gas temperature increases, the heat-transfer rate increases.

4.2. Effect of working fluid on heat-transfer rate (see Fig. 5)

In this experiment, the effect was measured for the CEOHP air-preheater with eight turns and 32 columns, and a hot-gas velocity of 3.3 m/s. Fig. 5 compares the heat-transfer rate experimental data to the heat-transfer rate predicted from the correlation [3] and the data [5]. If the working fluid changes from water to R123, the heat-transfer rate also increases, because the R123 has a lower latent heat of vaporization. It can be concluded that, if the working fluid is changed from water to R123, the heat-transfer rate increases.

4.3. Effect of hot-gas temperature on thermal effectiveness (see Fig. 6)

In this experiment, the CEOHP air-preheater has eight turns and 32 columns, and a hot-gas velocity of 3.3 m/s. Fig. 6 shows the effect of the hot-gas inlet-temperature on the effectiveness of the CEOHP air-preheater. It can be seen that, when the hot-gas inlet-temperature increases, the effectiveness also rises because the fresh-air outlet temperature also increases. Thus, the temperature difference between the inlet and outlet air also increases and the actual heat-transfer rate will be high. It can be concluded that, if the hot-gas inlet temperature increases, the effectiveness increases.

4.4. Effect of working fluid on thermal effectiveness (see Fig. 6)

In this experiment, the CEOHP air-preheater has eight turns and 32 columns, and a hot-gas velocity of 3.3 m/s. It can be seen in Fig. 6 that, if the working fluid changes
from water to R123, the effectiveness also increases because the R123 has a lower latent heat of vaporization.

5. Conclusion

In this study, we designed and built an experimental prototype to investigate the applicability of a CEOHP air-preheater as a waste-heat recovery device for a drying process. It can be concluded that:

- As the hot-gas temperature increases from 60 to 80 °C, the heat-transfer rate slightly rises.
- If the working fluid changes from water to R123, the heat-transfer rate slightly increases.
- The designed CEOHP air-preheater can achieve energy thrift.

Acknowledgements

The research has been supported generously by the Thailand Research Fund (Under Contract No. MRG4680023). The authors express their sincere appreciation for all of the support provided.

References


