Development of Heat Pipe Heat Exchanger

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Abstract

In this study, the gas-to-liquid heat pipe heat exchanger was fabricated and tested, and the thermal recovery capability and thermal performance of heat pipe heat exchangers were investigated. The device consists of 19 stainless steel-water heat pipes with an outside diameter of 6.2 mm, a wall thickness of 0.5 mm and a length of 300 mm. The condition for experiment was conducted where, hot gas and cooling waters enter at temperatures of 150-250 ℃ and 30℃, respectively. The flow rate of gas through the evaporator is 0.114-0.270 kg/min, while the flow rate of water through the condenser is 0.940 kg/min. The results showed that the maximum heat transfer rate was 445.5 W, and the maximum effectiveness was 0.609. The present research also compare between experimental and theoretical investigation with reference to the correlation in the literature.

Keywords: Heat pipe heat exchanger; Heat pipe; Thermal network

Introduction

In today's energy shortage crisis, how to save energy and reuse is a must pay attention subject, and waste heat emissions is one of the major sources of pollution. However, conventional heat exchangers used in industry often suffer corrosion and sediment from exhaust gases, leading to a reduction in heat recovery efficiency and equipment failure. Heat pipes can effectively transfer heat from one end to another, without additional force to drive heat transfer. Heat pipe heat exchanger is one of the practical applications of heat pipe, through the latent heat of vaporization to transfer heat over a long distance and small temperature difference characteristics. In the process of industrial manufacturing or processing, it needs great amounts of energy to heat raw materials. Afterwards, there will be about half of energy discharged in the form of waste heat in air. The waste heat can be commonly classified into high temperature (above 650 ℃), medium temperature (250 ℃ ~ 650 ℃) and low temperature (below 250 ℃), and low temperature accounts for a majority of waste heat emissions. As a result, if we can reuse heat energy efficiently, that will help energy conservation and carbon reduction a lot.

Heat exchangers made of heat pipes are one of the most effective pieces of equipment for waste heat recovery. A heat pipe heat exchanger consists of a number of these tubes arranged in rows. It operates with the evaporator section of the heat pipes in the high-temperature fluid stream and the condenser section in the low-temperature fluid stream. This study is aiming at the heat recovery capacity and performance analysis of mini heat pipe heat exchanger of low temperature exhaust gas (below 250 ℃), discharged from industrial waste heat. Also the theoretical analysis is carried out based on the relevant theoretical analysis in the literature.

2 Preparations

2.1 The specification for heat pipes

This experiment is for the heat pipe heat exchanger at low temperature industrial waste heat recovery below 250 ℃. Water have many benefits, such as it can be easy to get , and that it is the most widely used fluid, so we choose water to be the working fluid of the heat pipe in order to meet the operating temperature range. Since industrial waste heat contains gases such as hydrogen sulfide, sulfuric acid, dust and other pollutants and particles, the equipment gets easily corroded and damaged. This results as shortened equipment efficiency and life. We use 316L smooth tube made of stainless steel with high corrosion resistance and good strength to be the metal container with the following dimensions: 6.2mm OD, 0.5mm wall thickness and total length of 300mm. The capillary structure is mounted on the inner wall of the container with a layer of 200 mesh copper. The filling volume of the working fluid is injected into pores corresponding to the capillary structure, which is in the range of 1.4 - 1.5g. The metal container is filled with working fluid and then vacuumed to 1 x 10^{-2} torr, followed by secondary degassing to eliminate non-condensable gases. The completed heat pipe is shown in Figure 2.1.

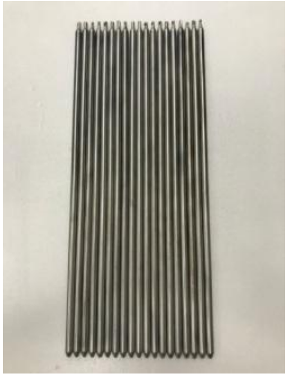


Fig. 2.1 Stainless steel - water heat pipe

2.2 Heat pipe heat exchanger

In this study, gas-liquid heat pipe heat exchanger was used to conduct experiments. The bottom evaporation end inputs the heated air, and the top condensation end inputs cooling water. The evaporator end of the device is a cuboid with a length of 200 mm, a width of 150 mm and a height of 260 mm. The condenser end of device is a cylinder with an outer diameter of 115 mm and a height of 140 mm, both made from steel. The design is shown in Figure 2.2. Total 19 Heat pipe can be installed inside the heat exchanger in orientation where, equilateral triangle staggered into a hexagonal shape. By the way, the total row number is 5, and the arrangement and spacing shown in Figure 2.3.

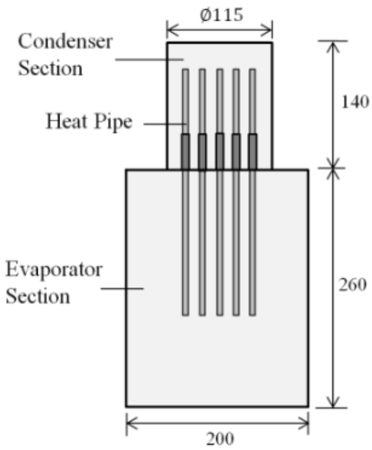


Fig. 2.2 Heat pipe heat exchanger internal map

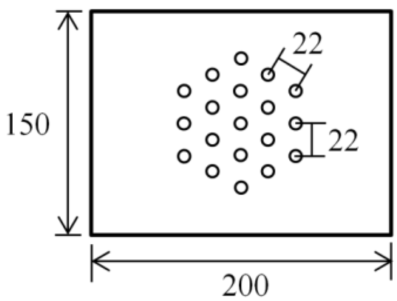


Fig. 2.3 Heat pipe arrangement

The test platform composed of a blower, hot air machine, anemometer, heat pipe heat exchanger, flow meter temperature acquisition system, and temperature water tank shown in Figure 2.4 and Figure 2.5. The environment air is blown into a stainless steel container with an inner diameter of 83.5 mm via a blower, and then flows through a hot air blower. After it will be heated by a hot air blower, hot air gets into the right side ($T_{h,in}$) of the heat exchanger. Then, after heat exchange with the heat pipe, the hot air will leave from the left side ($T_{h,out}$) of the heat exchanger. In other hand the condensation end flows from the left side ($T_{c,in}$) into the cooling water cycle to reduce the temperature and heat, then flows out to the right side ($T_{c,out}$). The water flow rate is controlled using a flow meter; also the anemometer is used to measure the air flow rate in the pipe. In the experiment, temperature of the fluid is read through the thermocouple wire connected to the temperature measurement module and recorded in the temperature acquisition system. The main temperature measurement points are respectively located at the points where hot gas and cooling water get in and out of the heat exchanger. Simultaneously, we will cover heat exchanger by cotton insulation to avoid heat loss and air convection interference.

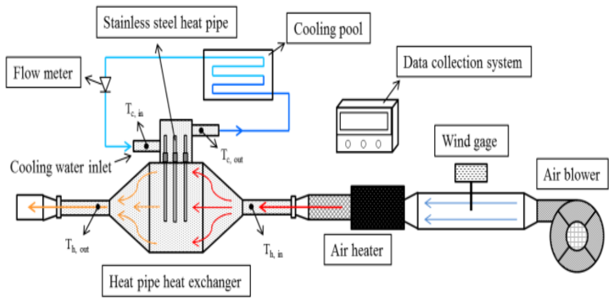


Fig. 2.4 Schematics of the experimental rig

Fig. 2.5 Picture of the heat pipe heat exchanger

2.3 Experimental methods and parameters

At the control panel, parameters are set for test, and then the temperature changes were observed and recorded during the experiment. The parameters are shown in Table 1. First, set a blower speed up for fix hot air mass flow rate; second keep hot air outlet temperature at 150 ℃. After fluid temperature reached steady state, we can keep going from 175 ℃ to 250 ℃, and so on. This study defines steady state of temperature as all temperature change is fluctuating not more than ± 1 ℃.

Table 1. Experimental parameters

Control parameters	
Heat pipe specifications	Metal containers:SS316L Outside diameter: 6.2mm Wall thickness: 0.5mm Total length: 300mm
Length of evaporator section and condenser section of heat pipe (mm)	Evaporator section 196 Condenser section 100
Quantity of heat pipes	19
Temperature of cooling water inlet (℃)	30
Mass flow rate of cooling water (kg/min)	0.94
Parameter variations	
Temperature of heating water inlet(℃)	150、175、200、225、250
Mass flow rate of heating water (kg/min) (rotating speed of air blower)	0.114 (20%) 0.192 (30%) 0.270 (40%)

2.4 Experimental error analysis

The experiment has errors due to instruments and human factors, thus it is necessary to carry out uncertainty analysis. The factors that can cause the experimental error in this study are; fluid mass flow rate and temperature. All of these experimental errors will lead to calculation errors for quantity of heat transfer and validation. The result about validity uncertainty can be given by the formula (2.1). [4]

$$w_\varepsilon = \left[\left(\frac{\partial \varepsilon}{\partial \dot{m}_h} w_{\dot{m}_h}\right)^2 + \left(\frac{\partial \varepsilon}{\partial \dot{m}_c} w_{\dot{m}_c}\right)^2 + \left(\frac{\partial \varepsilon}{\partial \Delta T_{in}} w_{\Delta T_{in}}\right)^2 + \left(\frac{\partial \varepsilon}{\partial \Delta T_c} w_{\Delta T_c}\right)\right]^{1/2} \quad (2.1)$$

which is the experimental error.

Through the experimental uncertainty analysis, we found the experimental measurement errors for hot gas mass flow rate is ± 2%, cooling water mass flow rate is ± 1% and import and export temperature is ± 1 °C respectively.

3 Theory

3.1 NTU- Effectiveness method

To evaluate the performance of a heat exchanger, it is usually analyzed by the Effectiveness (ε) and NTU methods. Effectiveness (ε) was defined as the ratio of the actual heat transfer (q) of the heat exchanger to the maximum possible ideal heat transfer (q_{max}) that would have occurred in a heat exchanger with an infinite surface. The exit temperature of the cold fluid would equal the inlet temperature of the hot fluid. It is defined as follows:

$$\varepsilon \equiv \frac{q}{q_{\max}} \quad (3.1)$$

The actual heat transfer quantity q is expressed as the exothermic quantity of hot fluids q_h or heat absorption quantity of cold fluids q_c, calculated as:

$$q_h = \dot{m}_h c_{p,h}\left(T_{h,in} - T_{h,out}\right) \quad (3.2)$$

$$q_c = \dot{m}_h c_{p,c}\left(T_{c,out} - T_{c,in}\right) \quad (3.3)$$

And the maximum possible ideal heat transfer quantity is as follows:

$$q_{\max} = C_{\min}\left(T_{h,in} - T_{c,in}\right) \quad (3.4)$$

In above formulas, it was founded that C_{min} was the smaller number between the heat capacity of hot fluids (C_h) and cold fluids (C_c). The heat capacity C was defined as the product of the fluid mass flow rate ṁ and the specific heat c_p. Therefore, validity can be ordered again from the formulas of (3.2), (3.3) and (3.4) as follows:

$$\varepsilon = \frac{C_h\left(T_{h,in} - T_{h,out}\right)}{C_{\min}\left(T_{h,in} - T_{c,in}\right)} \quad \text{if } \left(C_h > C_c\right) \quad (3.5)$$

or

$$\varepsilon = \frac{C_c\left(T_{c,out} - T_{c,in}\right)}{C_{\min}\left(T_{h,in} - T_{c,in}\right)} \quad \text{if } \left(C_c > C_h\right) \quad (3.6)$$

3.2 Heat pipe heat exchanger

This paper extended the definition of the performance formula for heat pipe heat exchanger by using theoretical calculation of the existing literature with the validity analysis. Faghri [5] analyzed heat pipe heat exchangers, first one considered as liquid-coupled indirect-transfer-type heat exchangers, which connects two direct-transfer-type heat exchangers through the heat pipe. Second is the direct-transfer-type heat exchanger exchanged heat with the working fluid for the hot side and the cold side respectively, as shown in Figure 3.1. In literature assessing the performance of heat pipe heat exchangers has been studied with good results [1, 2, 3]. In previous references we can see that the evaporation ends and condensation ends of heat pipe heat exchanger were divided into two direct-transfer-type heat exchangers and exchanged heat with heat pipes respectively. During heat exchange the working fluid in heat pipes was in the state of evaporated fluid gas mixture. When the fluid changed with each other, the working fluid temperature was almost constant and the specific heat became a considerable value. This makes working fluid heat content C_L increase dramatically. Therefore, it could be assumed that when hot fluids and cold fluids exchanged heat with working fluid respectively, the ratio of min heat capacityt to the max heat capacity would approach 0 ($C_{min}/C_{max} = C_r \approx 0$). Than, the formula of the heat exchanger effectiveness can be expressed as (3.7) [6]:

$$\varepsilon = 1 - \exp(-NTU) \tag{3.7}$$

NTU is an important dimensionless parameter in the analysis of a heat exchanger and is defined as follows:

$$NTU \equiv \frac{UA}{C_{\min}} \tag{3.8}$$

In the case of a heat exchanger with a single heat pipe, the effectiveness of both ends can be expressed as:

$$\varepsilon_{e,1} = 1 - \exp(-NTU_e) \tag{3.9}$$

$$\varepsilon_{c,1} = 1 - \exp(-NTU_c) \tag{3.10}$$

NTU_e and NTU_c above are as follows:

$$NTU_e \equiv \frac{U_e A_e}{C_e}; NTU_c \equiv \frac{U_c A_c}{C_c} \tag{3.11}$$

Furthermore, if the heat exchanger is a multi-row heat pipe, its effectiveness can be expressed by the following ε-NTU relationship [5]:

$$\varepsilon = \left[\left(\frac{1-\varepsilon_1 C_r}{1-\varepsilon_1}\right)^n - 1\right]\left[\left(\frac{1-\varepsilon_1 C_r}{1-\varepsilon_1}\right) - C_r\right]^{-1} \tag{3.12}$$

In the formula, n is the number of how many rows heat pipes are. Then, taking $C_r = 0$ into formula (3.12) can get the following relationship:

$$\varepsilon_{e,n} = 1 - (1-\varepsilon_{e,1})^n \tag{3.13}$$

$$\varepsilon_{c,n} = 1 - (1-\varepsilon_{c,1})^n \tag{3.14}$$

According to Kays& London [7], the overall effectiveness of heat pipe heat exchanger could be organized into following relationship from the formulas (3.9)、(3.10)、(3.13)、(3.14) :

If $C_e > C_c$ $\varepsilon_{e,n} = 1 - (1-\varepsilon_{e,1})^n$ (3.15)

If $C_c < C_e$ $\varepsilon_{e,n} = 1 - (1-\varepsilon_{e,1})^n$ (3.16)

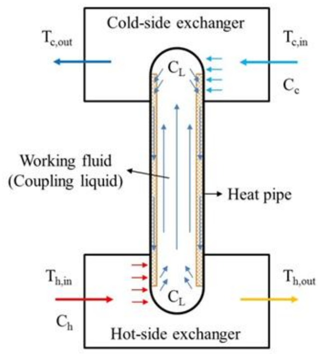


Fig 3.1 Schematics of heat pipe heat exchanger

4 Results and discussion

In the study, 19 stainless steel-water heat pipes were used to study the gas-liquid heat pipe heat exchanger. The hot air with different inlet temperature and mass flow rate was respectively put into the evaporation end. The condensation end was fixed with cooling water of inlet temperature and mass flow rate. However, the temperature changes of hot air and cooling water entering and exiting the heat pipe heat exchanger were recorded. The experimental results were further collated and calculated, and the thermal performance of the heat pipe heat exchanger under different operating parameters was analyzed. At the same time, the actual experimental values were compared with the theoretical values.

4.1 The temperature changes of cold and hot fluids

During the experiment, the hot air at the evaporation end of the heat pipe heat exchanger was 150, 175, 200, 225 and 250 ℃ at different mass flow rates of 0.114, 0.192 and 0.270 kg / min respectively. The condensing end maintained the inlet temperature of 30 ℃ and the mass flow rate of 0.940 kg / min. however, four thermocouple wires were used to measure the temperature change of the fluid, also the steady-state temperature data were collected to analyze the temperature change of the fluid.

Figure 4.1 to 4.3 showed the temperature changes of fluid over time in case of hot air in fixed mass flow rate and cooling water with constant inlet temperature and mass flow rate. The result shows that the inlet temperature of hot air and cooling water were all increasing gradually. The figure 4.4 indicate the relation chart between the distribution of temperature of fluid and mass flow rate with the inlet temperature of hot air 150 – 250 ℃. It revealed that when the mass flow rate of hot air increased, the inlet and outlet temperature of hot air and the difference between them would decrease. Nevertheless, the outlet temperature was higher while cooling water remains at 30 ℃.

It was founded from the results that if the mass flow rate of hot air increased, the thermal convection coefficient (U_e) rised accordingly. Which increase the heat transfer quantity of heat pipes, and that led to increase inlet temperature of cooling water. However, the reason for the decrease in temperature difference of hot air was because, increase of mass flow rate was larger than the rate of increase of heat transfer quantity. As a result, temperature difference still decreased as the heat transfer quantity increased.

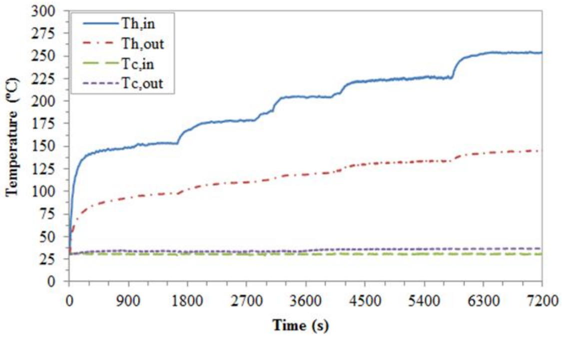


Fig 4.1when the hot air mass flow rate is 0.270 kg / min, the Fluid temperature changes with time.

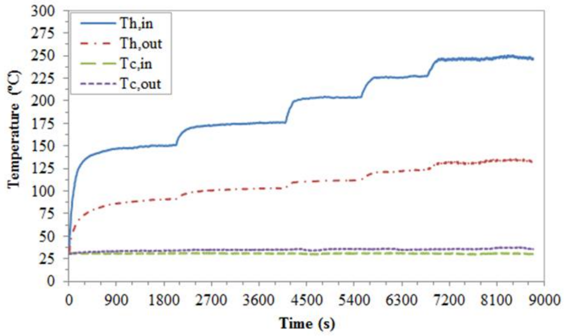


Fig 4.2 when the hot air mass flow rate is 0.192 kg / min, the Fluid temperature changes with time.

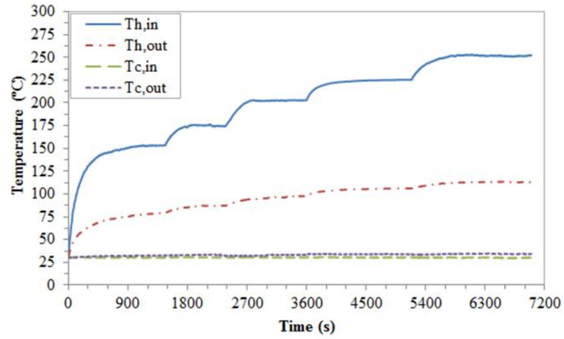
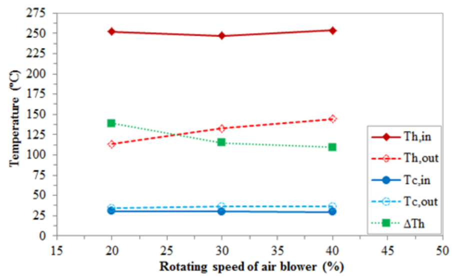
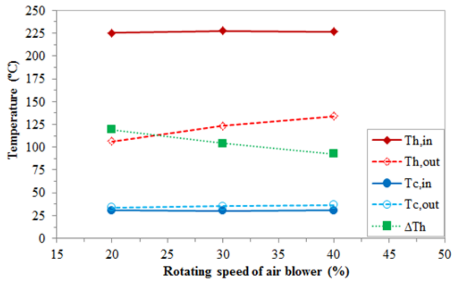


Fig 4.3 when The hot air mass flow rate is 0.114 kg / min, the fluid temperature changes with time.





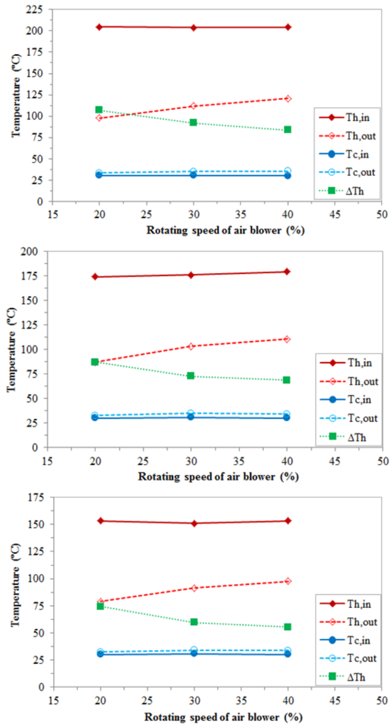


Fig 4.4 the hot air inlet temperature at 150 – 250 ℃, the fluid temperature distribution changes with the mass flow rate.

4.2 Heat transfer quantity

The outlet and inlet temperature data into the condenser (3.3) is calculated, where the actual heat transfer rate is obtained under various experimental parameter.

Figure 4.5 shows the effect of changing hot air mass flow rate to actual heat transfer rate. As the result shows, the actual heat transfer rate is increased when hot air temperature increased and the mass flow rate rise up. The heat transfer rate is the maximum at 445.5 W, when mass flow rate is 0.270 kg/min and inlet temperature is 250 ℃. The heat transfer rate is the minimum at 145 W, when the mass flow rate is 0.114 kg/min and inlet temperature is 150 ℃. Besides, from the curve of heat transfer, we can see that when the mass flow rate increases the increase of heat transfer rate tends to slow down. So the heat transfer rate should be at its maximum value. Figure 4.6 shows the error analysis on heat transfer rate. It can be found that the error between the heat transfer out from hot air (Q_{in}) and the absorbed by the cooling water (Q_{out}) is within 10%, which is acceptable.

4.3 Effectiveness

The effectiveness is an important factor according to Heat Exchanger Heat Transfer. Figure 4.7 represents the effect of hot air mass flow rate change to effectiveness. The experimental results can be found that the increase in hot air mass flow rate decreases the effectiveness. The effectiveness is the maximum at 0.609 when, the mass flow rate is 0.117 kg/min and inlet temperature is 150 ℃. The effectiveness is the minimum at 0.410 when the mass flow rate is 0.270 kg/min and inlet temperature is 225 ℃. The Uncertainty analysis of effectiveness shows in Figure 4.8.

When the air mass flow rate increases the percentage error of effectiveness decreases. The uncertainty gets higher during the lower temperature condition and decrease during the higher temperature condition.

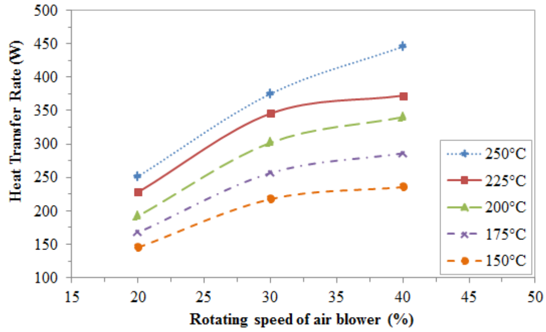


Fig 4.5 Power output according to inlet temperature and mass flow rate.

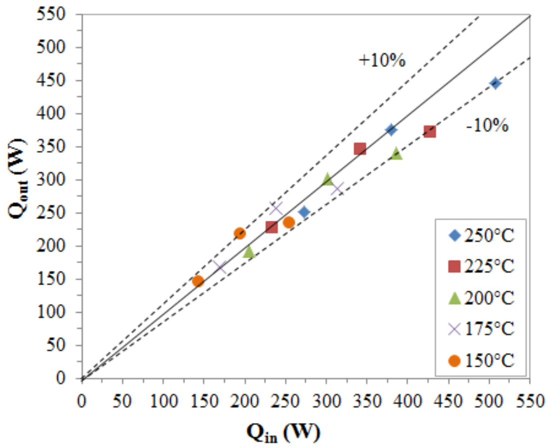


Fig 4.6 Relationship between heat release from hot air and heat absorption from cooling water

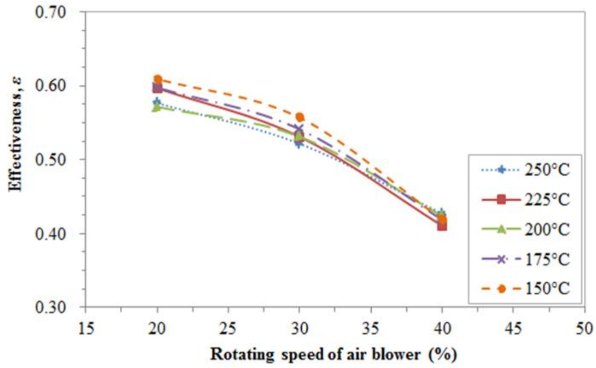


Fig 4.7 Effect of hot air mass flow on effectiveness

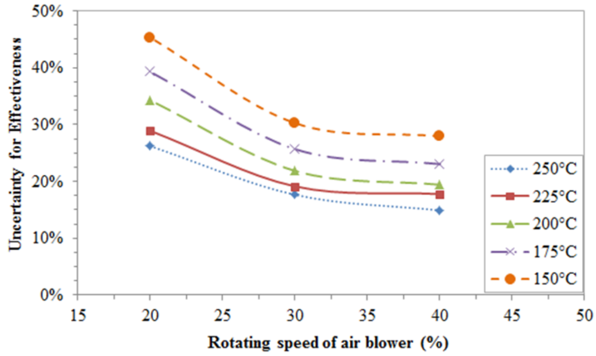


Fig 4.8Uncertainty Analysis of Validity

In accordance with previous discussion; as the mass flow rate increases, heat convection coefficient increases. This causes heat transfer rate to increase. Furthermore, heat transfer rate is proportional to the inlet temperature. The change in effectiveness is due to a span of hot air in heat exchanger. When the mass flow rate is lower, the velocity decreases and hot air in the heat exchanger stays longer. Therefore it can get higher effectiveness.

4.4 Theoretical and experimental value comparison and discussion

The comparison between the theoretical value and the experimental value in this section using the ε-NTU diagram is plotted using (3.16) and the effectiveness of the experimental result is compared with that of the NTU. Figure 4.9 shows the comparison between the ε-NTU and the experimental results. The vertical axis is the overall effectiveness and the horizontal axis is the hot side NTU_e. Since (3.16) contains the hot side and the cold side effectiveness, the cooling water is at constant mass flow rate, and its NTU_c is also a constant value. Therefore, the cold side effectiveness is assumed to be a fixed value and NTU_c calculated in this experiment is used to calculate the equation.

Figure 4.9 shows that the experimental effectiveness rise with the increase of NTU_e and the distribution is slightly lower than the theoretical curve. The theoretical analysis still has a good relationship with the experimental results and the experimental error is between 14.9 and 45.3%. The higher NTU_e value is closer to the theoretical curve. In addition, the heat transfer coefficient and heat transfer area of the heat pipe heat exchanger in this study are so small that it cannot transmit more than 0.25 heat transfer units, if the theoretical trends were fully developed, it can be found that when $C_r = 0$, $NTU_e = 1.5$. Also the overall effectiveness can be increased to nearly 100%, as shown in Figure 4.10. Therefore, the equation (3.15) and (2.22) could be used to plot more ε-NTU relationships. The study of the thermal performance of the exchanger can be directed towards NTU_e, NTU_c, the number of heat pipes and so on. Also the theoretical formula in the literature could be verified, which can be of great help to the design and performance prediction of the heat pipe heat exchanger.

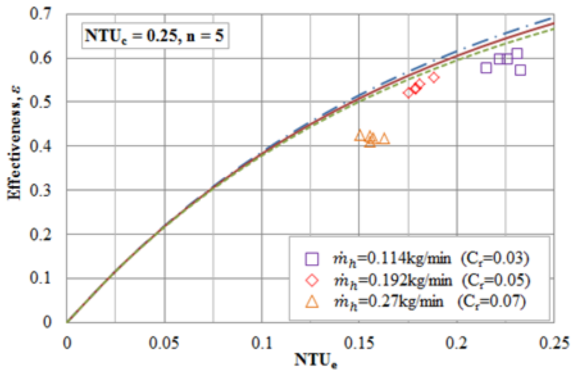


Fig 4.9 Comparison between the effectiveness of the heat exchanger and the NTU.

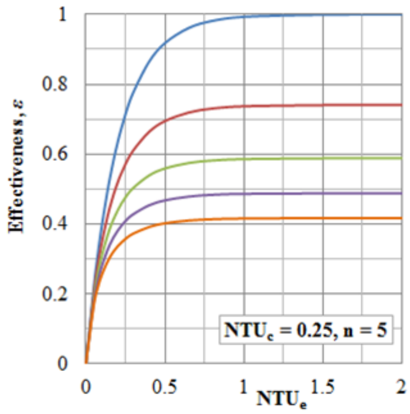


Fig 4.10 ε-NTU diagram of heat pipe heat exchanger

5. Conclusion

1. At a constant inlet cooling water temperature and mass flow rate, when the hot air mass flow rates increase, the hot gas and cooling water outlet temperatures increases at the same time.

2. With the increase of the mass flow rate of hot air, the heat convection coefficient is increased. Furthermore, the heat transfer increases with the increase of the heat transfer temperature. The maximum heat transfer in the experiment is 445.5 W, and the minimum Heat transfer is 145 W.

3. When the mass flow rate of hot air increases, the fluid velocity increases. This results in a short heat exchange time between the fluid and the heat pipe. As a result, the effectiveness decreases as the mass flow rate increases. The maximum effectiveness of the experimental results is 0.609, and the minimum is 0.410. Thus, at lower hot air inlet temperatures

and mass flow rates, the uncertainty of effectiveness is higher.

4. The error between ε-NTU plotted by the theoretical analysis and calculation of the literature and the experimental results, are between 14.9% and 45.3%

Nomenclature		
A	area	(m^2)
Q/q	Heat transfer rate	(W)
ΔT	Temperature difference	(℃)
ε	Effectiveness	
T	Temperature	(℃)
$\dot{m}$	Mas flow rate	(kg/s)
c_p	Heat capacity rate	(J/kg·K)
C	Thermal element	(W/K)
C_r	Thermal element ratio	
U	Overall heat transfer coefficient	($W/m^2 \cdot K$)
NTU	Number of Transfer Units	
Subscripts		
h	Hot	
c	Cold	
e	evaporator	
in	inner	
out	outre	
min	absolute minimum	
max	absolute maximum	

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