Experimental Investigation of the Performance of a Wrap-Around Heat Pipe at Different Filling Ratios

Mustafa ZABUN[*], Harun DENİZLİ

[1]*Friterm Termik Cihazlar A.Ş. İstanbul, Turkey*
[2]*Friterm Termik Cihazlar A.Ş. İstanbul, Turkey*

Highlights

- This paper focuses on Wrap Around Heat Pipe performance optimization
- The data used in the study was obtained by experimental method.

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Abstract

Heat pipe technology emerges as a viable, efficient and environmentally sensitive technology for applications in efficient air conditioning plant designs. In this study, firstly, the performance of a grooved and non-grooved Wrap-around heat pipe (HP) filled with R134a at the same filling ratio has been experimentally compared. Then, the capacity of the grooved heat pipe (WHP) with a 3° inclination angle, whose performance is higher than the non-grooved WHP with smooth tubes, at different filling ratios was experimentally investigated. These filling ratios are Filling Ratio 1, Filling Ratio 2, Filling Ratio 3 and Filling Ratio 4, respectively. The WHP dehumidifier unit with grooved tubes with a diameter of 7.94 mm and 3° inclination angle with 60-circuits was used. In the experiment, the ambient temperature was kept constant at 25°C to heat the evaporator part of the WHP, and a cooling coil was inserted between the WHP to achieve the dehumidification of the pre-cooled air. The results obtained are compared with the capacity and efficiency of the WHP for different filling ratios. In line with the results obtained, it was observed that the best WHP efficiency and the highest WHP capacity were also achieved at Filling Ratio 1. In addition, increasing WHP capacity has been found to save 14% on the energy needs for cooling.

1. INTRODUCTION

With an exponential increase in global energy demand, it is evidently clear that energy resources should be used in a discreet and responsible manner in order to conserve them for future generations. In process industry and building sector, a significant amount of energy is wasted through the discharge streams.[1] This waste should be energy efficient, safe and easy to use so that it can be preferred in industrial applications. Nowadays, heat pipe heat recovery units are preferred more than traditional systems, considering system efficiency, ease of production and cost. The main reason for this is mainly due to passive works and developments in production technologies. [2]

Heat pipes are two-phase thermal products with strong thermal conductivity even at low temperature differences. Heat is transferred from the evaporator to the condenser by a small amount of working fluid such as water, methanol, R134a or sodium in the form of latent heat. Many different types of heat pipes are presented in the literature and used in the HVAC industry [3].

People living in hot and humid climates need large amounts of energy to ventilate their living spaces. In such climates, the moisture load is quite high and an external moisture content of up to about 25 g/kg is common. Cooling systems must be sized to cope with these high latent heat loads. Heat pipes have been used in these cooling applications for a long time and have provided significant energy savings [4]. The process of dehumidifying the outside air takes place as follows; It is the process of super cooling to remove humidity from the environmental air and then reheating it by feeding it to a suitable temperature. This process consumes energy and uses desiccant heat pipes wrapped around the primary cooling coil (evaporator side). The heat pipe pre-cools the air before it reaches the cooling coil and is then reheated in the secondary coil (condenser side) after exiting the cooling coil. This combination of free cooling and free heating removes a significant amount of humidity.

The integration of heat pipes into heat exchangers could obviously maximize heat transfer rates while minimizing manufacturing cost, weight, size and overall thermal resistance [5]. In the past decades, heat pipe heat exchangers have been widely used in electronic refrigeration [6], air conditioning systems [7] and waste heat recovery [8].

*Correspondingauthor, mustafazabun@friterm.com

In order to increase the thermal performance of the heat pipes, both experimental and theoretical studies have been carried out by considering the design and operational parameters.. Mahdavi et al. experimentally investigated the thermal performance of a curved copper-water heat pipe at various filling rates and inlet temperatures. It has been conluded that the experimental results were in agreement with the previously developed mathematical model [9].
Guo et al.[10] conducted a series of experiments in order to evaluate the thermal performance of a wraparound heat pipe charged with R134a at different operating conditions, including various heat loads, filling ratios, pipe diameters, inclination angles and coolant temperatures. According to the experimental results, an optimal filling ratio for the heat pipe with the best performance existed between 50% and 60% and even at a proper filling ratio, a smaller inclination angle (θ less than 15◦) can result in very limited liquid in the upper evaporator tube, thereby reducing the thermal performance of the wrap- around heat pipe.

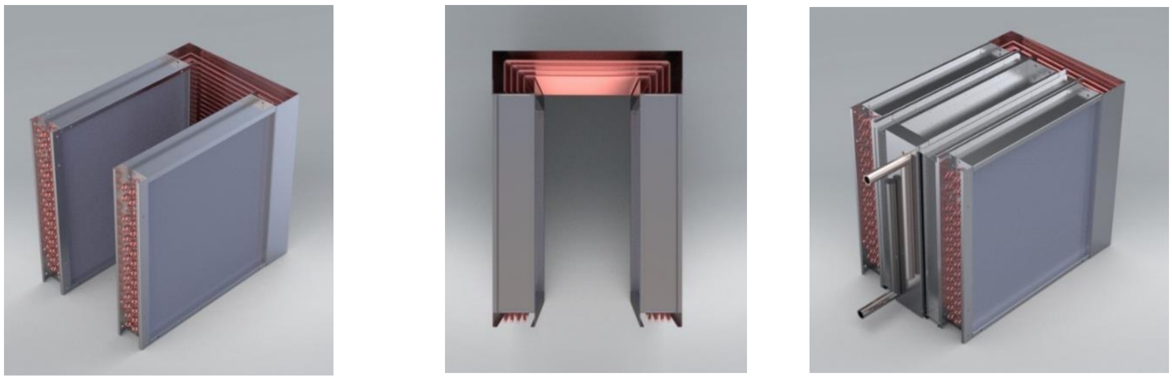

Figure 1. *Wrap-around heat pipe heat recovery units*

Considering the above, this paper aims to examine the thermal performance of the wrap-around heat pipe heat recovery (WHP) products in the product range of Friterm Thermal Devices Inc. at different filling ratios. In line with the results obtained, the optimum filling ratio was determined.

2. EXPERIMENTAL METHOD

The experimental study was carried out on a heat pipe dehumidification unit (WHP) produced by Friterm Thermal Devices Inc.

2.1. The Wrap-Around Heat Pipe (WHP) Design

The WHP design shown in Figure 2. WHP is made of 5/16" diameter grooved and smooth pipe with 0.3 mm wall thickness inside. It consists of 32 pipes, 4 rows and 32 circuits. Fin material is Aluminum and wall thickness is 0.12 mm. Fin length is 750 mm.
The heat exchanger used for cooling the coil section has a pipe diameter of 15 mm and a wall thickness of 0.4 mm. The coil consists of 20 tubes, 4 rows and 8 circuits. The fin material of the cooling coil is aluminum and is 0.10 mm thick. Coil fin length is 745 mm.

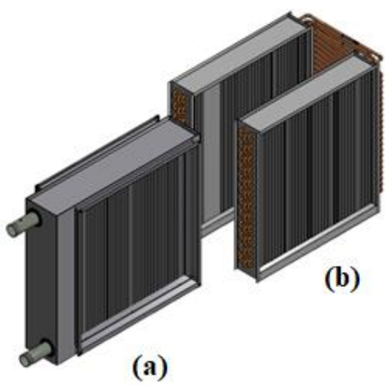

Figure 2. View of (a) coiling coil and (b) Wrap-around heat pipe

2.2. Experimental Setup

The experiments were made in the air-conditioning room of Friterm Thermal Devices Inc. Laboratories. The room was built in 2007. As can be seen in Figure 3 the test setup of the air-conditioning chamber consists of 4 main sections

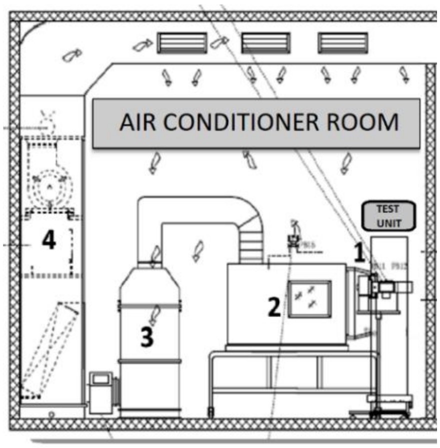

***Figure 3.** Air Conditioner Room System Design*

The first section is the tunnel part where the product to be tested is connected to the assembly. Here, some data is taken to determine the capacity of the product at the inlet and outlet. To verify the sensors, measurements were taken from the same points again with the precision hand probe. These values are:

- Evaporator Inlet Temperature
- Evaporator Inlet Absolute and Relative Humidity
- Evaporator Outlet / Cooling Coil Inlet Temperature
- Evaporator Outlet / Cooling Coil Inlet Humidity
- Cooling Coil Outlet / Condenser Inlet Temperature
- Cooling Coil Outlet / Condenser Inlet Humidity
- Condenser Outlet Temperature
- Condenser Outlet Absolute and Relative Humidity
- Air Flow

The 1-dimensional model of the measured points can be seen in Figure 4.

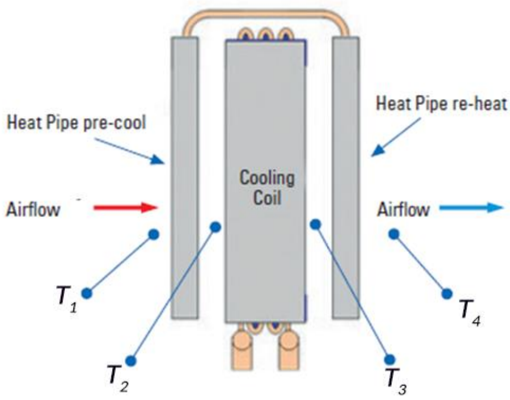

***Figure 4.** Measurement Points on the 1D Heat Pipe Model*

In the second section, there is a sight glass and a humidity eliminator. Here, this system is used to prevent water droplets reaching the fan. Part 3 is the part where the flow regulation nozzles are located. There are

4 nozzles with different cross sections. These nozzles are used to regulate the air flow of the centrifugal fan blower. These nozzles of different sizes are used to set different velocities. In the 4th part, there are centrifugal fan and heaters. The task of the heaters here is to keep the room in thermal balance by heating the air up to a limit in consistence with the capacity of the product installed in the test setup. The heated air is supplied to the room and the room remains at a thermal balance.

All filling ratios were achieved under the test conditions given in Table 1.

Table 1. *Test Conditions for all filling ratios*

Air Inlet Temperature	K	298,15
Air Relative Humanity	%	80-76
Air Mass Flow	m^3/h	2500-2506
Water Inlet Temperature	K	284.15-284.7
Water Outlet Temperature	K	289.15-290.15
Water Mass Flow	m^3/h	2.7

The properties of the R134a refrigerant at 298,15 K are given in Table 2.

Table 2. *The properties of the R134a refrigerant*

Density	[kg/m^3]	1206.07
P_{sat}	[MPa]	0.66538
C_p	[kj/kgK]	1.4246
Viscosity	[Pas]	0.00019489
Enthalpy	[kj/kg]	234.55

The temperature of the air leaving the evaporator section is measured at a total of nine points with thermocouples placed on 3 thin wires stretched across the flow direction. In addition, the relative humidity of the air leaving the evaporator was measured by 2 different devices. The main reason for this is that 2 different devices are aimed to verify each other. Figure 5 shows the location of both the thermocouples and the relative humidity sensors. This is also valid for the coil outlet section, that is, the condenser inlet section. In all experiments, a total of 18 T-type thermocouples and 2 constant relative humidity sensors and 1 relative humidity sensor with handy probe were used on both sides.

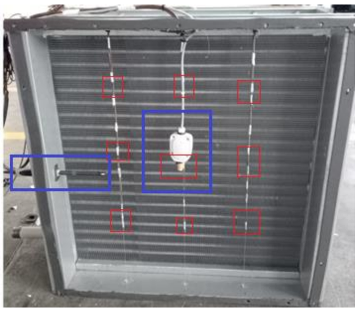

Figure 5. *Locations of thermocouple and RH sensors*

Figure 6. *View of the insulated adiabatic region*

The temperature of the adiabatic zone is assumed as the average of the air entering and leaving the WHP. The adiabatic side is insulated with insulating materials to reduce heat transfer from and to the adiabatic section.

3. NUMERICAL METHOD

T_1 is the temperature of the air entering the evaporator section. It is measured with a thermocouple placed in front of the WHP.

The average outlet temperature of the evaporator section is calculated as in Eq.1;

$$T_2 = \frac{T_{T1} + T_{T2} + T_{T3} + T_{T4} + T_{T5} + T_{T6} + T_{T7} + T_{T8} + T_{T9}}{9} \quad (1)$$

Here $T_{T1} \ldots T_{T9}$ refers to the thermocouples used in the experimental setup. The average outlet temperature of the coil section is calculated as in Eq.2;

$$T_3 = \frac{T_{T10} + T_{T11} + T_{T12} + T_{T13} + T_{T14} + T_{T15} + T_{T16} + T_{T17} + T_{T18}}{9} \quad (2)$$

Here $T_{T10} \ldots T_{T18}$ refers to the thermocouples used in the experimental setup.

T_4 is the temperature of the air leaving the condenser section. It was measured with the thermocouple in the mixing cabinet.

An accurate experimental determination of the thermal performance of the WHP requires accurate measurements of the evaporator and condenser temperatures as well as the heat transferred in between.

The heat transferred from air to the evaporator section is calculated as follow;

$$Q_{evap} = \dot{m}_{air} C_p \left(T_1 + T_2 \right) \quad (3)$$

The heat transferred from from the condenser section to the air stream is calculated as follow;

$$Q_{kon} = \dot{m}_{air} C_p \left(T_3 + T_4 \right) \quad (4)$$

The heat transfer rates of the evaporator and condenser sections are expected to be equal to each other. Therefore, the actual heat transfer rate can be calculated from the following equation:

$$Q_{evap} = Q_{kon} \quad (5)$$

In addition, the capacity of coil section is calculated as follow;

$$Q_{coil} = \dot{m}_{air} C_p \left(T_2 + T_3 \right) \quad (6)$$

The same mass of air flows across both evaporator and condenser sections of the WHP. If an adiabatic operation is considered, heat transferred between evaporator and condenser sections is assumed to be equal. In this case, energy balance determines that the temperature drop in the air stream across the evaporator section (precool) is equal to the temperature rise in the air stream across the condenser section (reheat). In some cases of high humidity, condensation within the air stream occurs in the evaporator section.

When this occurs, the evaporator section experiences both sensible and latent heat transfer while the condenser section would only undergo sensible heat transfer. Hence, the temperature rise across the condenser section (reheat) would be greater than the temperature drops across the evaporator section (precool).

This will affect the performance of the WHP. The sensible heat, latent heat or total energy effectiveness ε of the WHP is given as:

$$\varepsilon = \frac{\dot{m}_{air,ei} \left(T_1 - T_2 \right)}{\dot{m}_{air,\min} \left(T_1 - T_3 \right)} \quad (7)$$

Evaporator mass flow rate is represented by $\dot{m}_{ei}$, and $\dot{m}_{min}$ is the smallest of either of the two air streams.

When both air flow rates are equal, effectiveness of the WHP is as follows:

$$\varepsilon = \frac{T_1 - T_2}{T_1 - T_3} = \frac{T_4 - T_3}{T_1 - T_3} \tag{8}$$

3.1. The Uncertainty analysis

The expanded uncertainty was obtained by combining the individual standard uncertainties. Assuming that the desired parameter y is dependent on the experimental variables, x_1, x_2, x_3, …, each of which fluctuates in a random and independent way. That is, y is a function of x_1, x_2, x_3, …, and can be written as:

$$y = f(x_1, x_2, x_3, ...) \tag{9}$$

Accordingly, the overall uncertainty of parameter y is calculated as follows:

$$u_y = \sqrt{\left(\frac{\partial y}{\partial x_1} u_{x_1}\right) + \left(\frac{\partial y}{\partial x_2} u_{x_2}\right) + \left(\frac{\partial y}{\partial x_3} u_{x_3}\right) + ...} \tag{10}$$

Where, u_{x1} is the uncertainty of each parameter.
The uncertainty of the capacity can be estimated as follow;

$$Q = \sqrt{\left(\frac{\Delta \dot{m}}{\dot{m}}\right)^2 + \left(\frac{\Delta(\Delta T)}{\Delta T}\right)^2} \tag{11}$$

The type T thermocouples used in this study have the highest accuracy of all base metal thermocouples at ±1°C or ±0.75%, whichever is greater. Humidity sensor accuracy used in all tests is ±2 %RH. Also, the accuracy of the nozzle system in which the air velocity is measured is ± (0.1 m/s ±1.5%).

3. RESULTS

In order to evaluate the thermal performance of the WHP, the temperature and pressure of the working fluid, the wall temperature profiles in the evaporator and condenser, and the thermal resistance are compared under the same experimental conditions.. Filling ratio is defined as the ratio of working fluid volume to the total inside volume of the heat pipe. Contrary to the experiments by Guo et al[10]. using one circuit, this study was carried out on a finished product with 60 circuits.

In line with the experimental data obtained, it has been observed that the dehumidification process takes place as on the psychometric diagram. Theoretically, rate of precooling should be equal to the rate of reheating. Looking at Figure 7 and Figure 8, the absolute humidity difference between the dehumidification process start and end points can be seen.

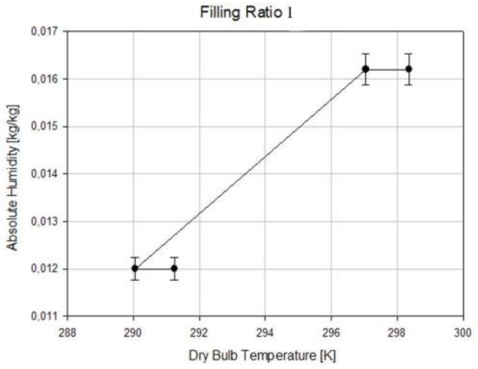

Figure 7. Dehumidification process in psychometric chart at Filling Ratio 1

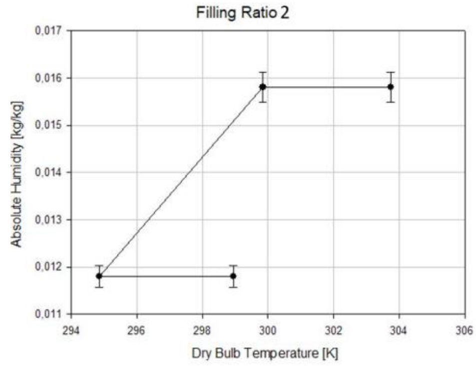

Figure 8. *Dehumidification process in psychometric chart at Filling Ratio 2*

Firstly, wrap-around heat pipe tests with grooved pipes and smooth pipes were carried out at Filling Ratio 2. In line with the results obtained, when the wrap-around capacities with smooth pipes and grooved pipes have been compared and as a result it has been seen that the capacity has increased by 46.97%. (Figure.7). While the R134a refrigerant was expected to perform week due to the friction flow in the grooved pipe, contrarily,the increase in heat transfer eliminated this expectation.

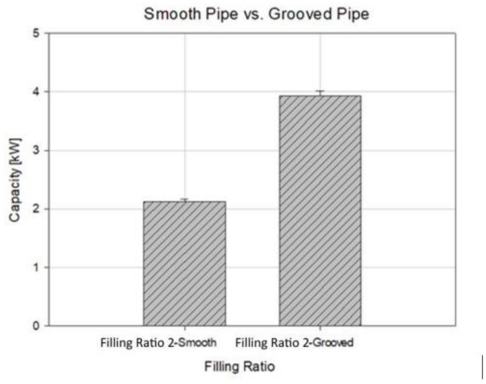

Figure 9. *Comparison of grooved and smooth tube wrap-around heat pipe.*

Generally, low filling ratio increase the possibility of drying in the evaporator section, and high fill rates inhibit fluid movement in the heat pipe. Looking at Figure 10, it could be seen that the highest capacity is obtained at Filling Ratio 2 and the lowest capacity at Filling Ratio 1. This means that the R134a movement is restricted at Filling Ratio 1. In addition, the fact that the circuit angle is 3° can be shown as a factor in this situation.[10]

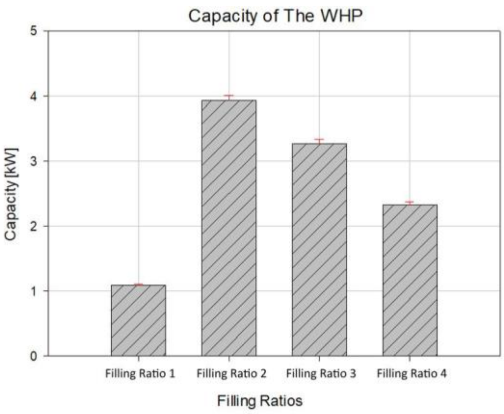

Figure 10. *Capacity of the WHP at different filling ratios*

Figure.11 shows the efficiency of the wrap-around heat pipe at different filling ratios. It is a known fact that the capacity of the coiled heat pipe is directly proportional to the efficiency. The best efficiency of the wrap-around heat pipe is at Filling Ratio 2, at Filling Ratio 3, Filling Ratio 4 and Filling Ratio 1, the WHP has performed from high to lower respectively

Figure 11. *Efficiency of the WHP at different filling ratios*

Figure 12 shows the capacity of the cooling coil, which undertakes the dehumidification process, at different filling ratios. The dehumidification process occurs when the incoming air is cooled lower than the dew point temperature and condensed on the coil. Then, this cooled air is reheated in the condenser section and the dehumidification process is completed. By increasing the efficiency of the wrap-around heat pipe, a 15% saving was achieved in the capacity of the coil used for cooling.

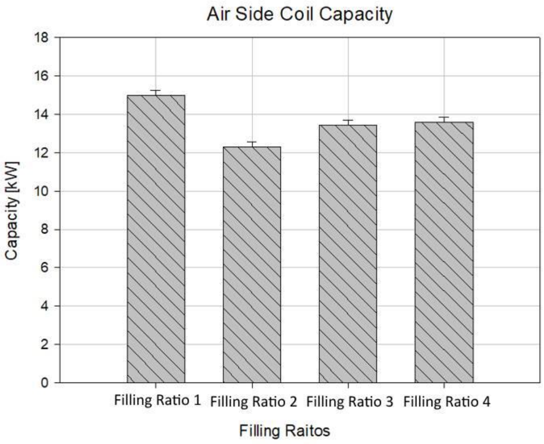

Figure 12. *Air side coil capacity at different filling ratios*

4. CONCLUSION

Experimental study was carried out to examine the effect of filling ratio on the thermal performance of WHP's. The experimental setup was used to optimize the thermal performance of a fixed geometry WHP. The difference of this study from other studies in the literature was that WHP has been investigated on the basis of a multi-circuit product, not on a single circuit, and the filling ratio was optimized.

In the first stage, the difference between grooved pipe and smooth pipe was examined. As can be seen in Figure.9 grooved pipe has higher heat transfer coefficient than straight pipe. The changes in heat transfer coefficient can be explained to be caused by the fact that the inner surface of grooved tube was greater than smooth pipe.

In the second stage, results were obtained at the same air flow rates for different filling ratios. The numerical results indicate that certain filling ratios will optimize the performance of the wrap-around heat pipe. It can be seen in Figure.10, as a result of our experiments, the performance of the wrap-around heat pipe was found to be maximum at filling Ratio 2.

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