PERFORMANCE IMPROVEMENT OF THE HEAT RECOVERY UNIT WITH SEQUENTIAL TYPE HEAT PIPES USING TiO_2 NANOFLUID

Ahmet ÖZTÜRK[1], Mehmet ÖZALP[2], Adnan SÖZEN[3,*], Metin GÜRÜ[4]

[1] Artvin Çoruh University, Artvin Vocational School, Machine Part, 08100 Artvin, Turkey

[2] Karabük University, Engineering Faculty, Mechanical Engineering78050 Karabük, Turkey

[3] Gazi University, Technology Faculty, Energy Systems Engineering, 06500 Ankara, Turkey

[4] Gazi University, Engineering Faculty, Chemical Engineering, 06570 Ankara, Turkey

[*)] Corresponding author: e-mail: asozen@gazi.edu.tr, Tel: +90 (312) 202 89 42

This paper deals with the improvement of thermal performance of the heat recovery system in air-to-air unit by using a nanofluid of TiO_2 (titanium dioxide) particles and distilled water. The experimental setup equipped with 15 copper pipes of a 1000 mm length, 10.5 mm inner diameter, and 12 mm outer diameter was used. The evaporator section consists of 450 mm of heat pipes, the condenser section is 400 mm, and the adiabatic section is 150 mm. In experimental studies, 33% of the evaporator volumes of heat pipes were filled with working fluids. Experiments were carried out at temperatures between 25ºC and 90ºC by using five different cooling air flows (40 g/s, 42 g/s, 45 g/s, 61 g/s, and 84 g/s), and two different heating powers (3 kW and 6 kW) for the evaporation section, to determine heat removed from the condensation section. Trials were performed for distilled water and nanofluid respectively, and the results were compared with each other. Results revealed that a 50% recovery in the thermal performance of the heat pipe heat recovery system was achieved in the design using TiO_2 nanofluid as the working liquid, at a heating power of 3 kW, air velocity of 2.03 m/s and air flow of 84 g/s.

Keywords: *Heat pipe, nanofluid, TiO_2, heat recovery unit*

1. Introduction

Today, due to increases in global warming caused by developing technology, the importance of environmentally friendly, energy-saving heat recovery units is increasing in the heating–cooling and air-conditioning sectors.

Heat pipes have been used in many areas since 1942, including in cooling space vehicles, as well as electrical and electronic devices, and in medicine in various ways, including for controlling human body temperature, and using as heat exchangers in other heat recovery systems [1]. The heat pipe is a highly effective device for high-temperature heat transfers over long distances with its easily controllable and simple structure that provides exceptional flexibility with infinitesimal temperature variations [2]. The fundamental of a heat pipe is based on the thermosyphon mechanism, also known as the "Perkin Pipe" and it has the same working principle as a two-phase thermosyphon [3]. As shown schematically in Figure 1, when heat is applied to a fluid flowing in a vertical, vacuumed, and impermeable pipe, then fluid starts evaporating; and the generated vapor moves towards the top of the pipe. Vapor reaching the top of the pipe condenses by releasing its heat outside the pipe and then becomes liquid. The fluid in the liquid phase reaches the bottom of the pipe due to gravity, thus the cycle is completed. The bottom of the pipe is the evaporator section where heat is removed from outside due to evaporation, while the top of the pipe is the condenser section where heat is released outside due to the condensation [4]. Zhu et al. [5] investigated the heat transfer performance of a closed heat pipe using distilled water (DW) and pure acetone (PA) at various mixing ratios (DW / PA ratio = 13:1, 4:1, 1:1, 1:4 and 1:13), as well as different heat inputs (10–100 W). In Riffat and Gan's [6] study carried out to determine the effectiveness of heat pipe heat recovery, a heat pipe heat recovery unit was developed to recover the low-temperature exhaust heat of air-conditioning units for heating fresh air in commercial buildings and houses. Numerous experiments were took placed and they determined that effectiveness of the system varied between 7 to 45% depending on working parameters (exhaust and fresh air input-output temperatures, as well as exhaust and fresh air flows). Wang et al. [7] revealed that high-temperature, special-shaped heat pipe (HTSSHP) coupling flat-plate heat pipe (FHP), and cylindrical heat pipes (CHPs) as the thermal receiver and heat transfer unit, was the significant component in a solar chemical reactor. They experimentally investigated and analyzed the isothermal performance and thermal resistance variation of HTSSHP as startup characteristics. In tests performed, heat transfer limits were not encountered. It was observed that general thermal resistance of HTSSHP decreased with increasing operating temperature from 0.12 to 0.19 ºC/W, similar to the same magnitude in a typical heat pipe. Moreover, they observed that various heat inputs and cooling rates had important effects on the thermal resistance in the cooling side.

Zhu et al. [8] designed an air-to-air heat exchanger for heat recovery in houses by using sequential micro heat pipes and produced them to test under normal temperature operating conditions. The heat exchanger has 4 rows and there are 12 heat pipes in each row. Experiments were done under both summer and winter conditions. In summer conditions, the indoor temperature was kept at 24ºC, while the outdoor temperature varied between 27ºC and 40ºC. In winter conditions, the indoor temperature was kept at 20ºC while the outdoor temperature varied between 9ºC and 14ºC. The air velocity range was between 0.64 and 1.50 m/s. According to the test results, the difference between indoor and outdoor air temperature, as well as velocity difference, significantly affected the efficiency of the heat recovery unit. The results of experiments revealed that the designed heat exchanger improved energy and energy recovery and had great potential for energy savings in large buildings. Sadeghinezhad et al. [9] experimentally examined the thermal performance of a sintered-wick heat pipe. They obtained results using graphene and distilled water for four different heating powers (20, 40, 60 and 80 W) by holding the heat pipe at two different positions (at 0 and 60 degrees with the horizontal). They observed a maximum reduction of 48.4% in the thermal resistance when using nanofluid instead of distilled water. When compared with 0º horizontal tilt angle, the improvement rate in the thermal conductivity was found to be 37.2% at the maximum tilt angle of 60º and the heating power of 60 W. Noie [10] investigated the thermal performance of an air-to-air thermosyphon heat exchanger by using the Number of Transfer Units (NTU) method. Orr et al. [11]-based on the knowledge that internal combustion engines are not efficient enough to convert chemical energy into mechanical energy—conducted studies on recovering the waste heat released from the cooling and exhaust, rather than increasing the engine efficiency. They found that heat pipes were useful for this recovery, with their advantages of being passive, silent, scalable and stable. Gedik et al. [12] investigated the usefulness of heat recovery system applications by using the heat pipe bundle charged with R134a and R410A working fluids, and also the thermal performance of a gravity-assisted heat recovery system. Using an experimental setup, they conducted test series using the flue gas temperature (75, 100, 125, 150 and 175ºC), flue gas velocity (1.0, 1.5, 2.0 and 2.5 m/s), and cooling water flow rates (1, 2, 3 and 4 l/min). The overall effectiveness values of the heat pipe heat recovery system varied with flue gas velocity, as expected. Experiments revealed that the heat pipe effectiveness values averagely changed from 35.6% to 57.7% for both R134A and R410A working fluids. Quantitatively, it was found that the R134a working fluid was 14% more effective than the R410A working fluid in the heat pipe bundle system.

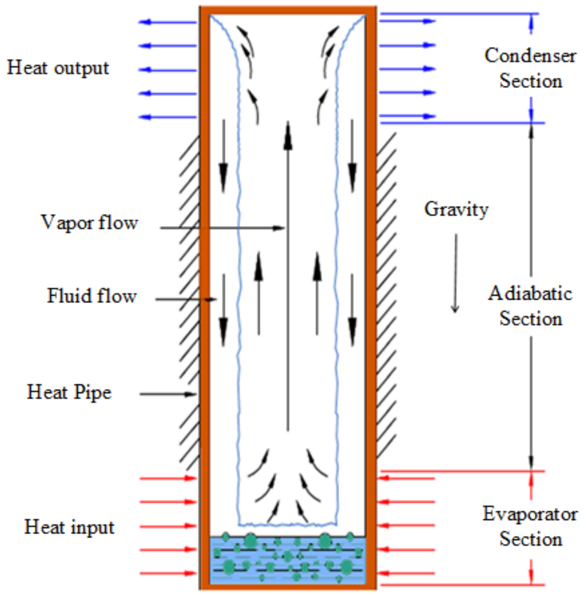

Figure 1. Heat pipe and its sections

Air to air heat exchangers are widely used equipments in a variety of applications. Particularly considering the type used for fresh air preheating by using waste heat, heat pipe including systems are more preferred for increasing the fresh air inlet temperature with transferred heat to fresh air. In order to be operated these systems under low waste heat temperatures, the working fluid inside the heat pipe must be evaporated at low temperatures. In this study, TiO_2 nanoparticles-including nanofluid, which evaporates at lower temperatures than deionized water, was used to solve this problem. So, both heat pipe performance and the thermal performance of the heat recovery unit could be improved.

2. Air-To-Air Heat Recovery Units

Air-to-air heat recovery units can be mainly divided into four groups: plate heat recovery units, wheel-type heat recovery units, battery system heat recovery units, and heat pipe heat recovery units.

2.1. Heat Pipe Heat Recovery Units

Heat pipes heat recovery units resemble wing tube heat exchangers. Pipes are individually independent from each other or combined groups. Phase-change fluids are injected into the individual pipes or common circuits. These fluids absorb or emit heat during phase change, or change their phases when they absorb or emit heat. Heat transfer and heat recovery is achieved thanks to these thermophysical properties and structures. Their lengths are separated into two equal lengths by a separator placed in the middle. Exhaust passes through one side of the separation while fresh air passes through the other side of the separation. The separation is only for preventing and directing the mixing of two air flows; it does not divide the flow in pipes or circuits. Heat exchanger fins are used to increase the surface and efficiency of heat transfer; they are not functional [13].

3. Experimental Method

3.1. Preparation of Nanofluid

The working fluid increases the amount of heat transfer by suspending nanoparticles in the prepared liquid. Titanium dioxide (TiO_2) is a very useful semiconducting transition metal oxide and has many positive features, such as low cost, ease of use, no toxicity, and resistance to photochemical and chemical erosion. Properties of TiO_2 are significantly dependent on its crystal phase. TiO_2 particles were made to be nanoparticles with a size of 14 to 17 nm by processing and purifying them in a trademark of SPEX 8000 high energy impacted ball milling. They have a size of 5 to 50 nm in normal distribution. Nanofluid contains 2 % of nanoparticles by mass. 0.2% Triton X-100 was added into the solution as a surfactant to prevent formation of coagulation and precipitation. TiO_2 nanoparticles were treated in an ultrasonic bath with water for five hours to homogeneously disperse them in distilled water, resulting in the required nanofluid. The container was hermetically closed to avoid evaporation of liquid during the preparation of the nanofluid.

3.2. Heat Exchanger

Figure 2 shows the designed heat exchanger bundle consisting of 15 heat pipes. Each heat pipe was made of copper and all of them were manufactured independently of each other. The copper pipes have a 12 mm outer diameter, a 10.5 mm inner diameter and a 1000 mm length. Heat pipes were manufactured as follows: the evaporator section is 450 mm, the adiabatic zone is 150 mm, and the condenser zone is 400 mm. Ten K-type thermocouples were placed on five heat pipes randomly selected from the heat exchanger bundle. Five out of ten K-type thermocouples were positioned in the condenser section and the remaining five thermocouples were positioned in the evaporator section; each of five thermocouples was placed on the same heat pipe (see Fig.2).

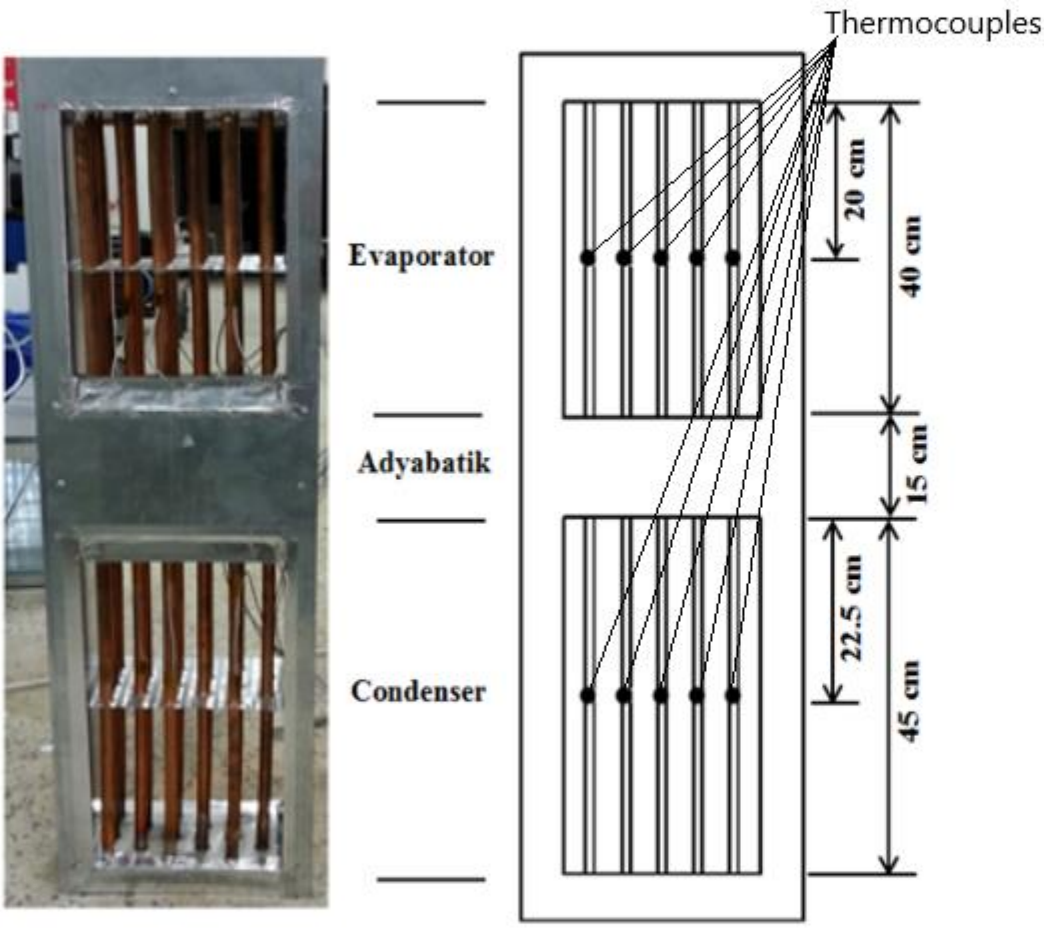

Figure 2. The designed heat exchanger bundle

The heat transfer mechanism takes place from the walls of the heat pipe towards the fluid. As it evaporates, the fluid is transferred to the condenser section from the evaporator section. The transferred heat is then transferred to a cold source. Vapor in the condenser section condenses and returns to the evaporator section through pipe walls due to the gravity. This process periodically occurs when heat input to the system is continuous. Briefly, it is possible to say that the heat is transferred from one section to another by utilizing the phase change of the working fluid.

3.3. Experimental Study

Figure 3 shows a schematic view of the experimental setup, while Figure 4 shows a photograph of the setup where experiments were carried out. The test regions were correlated

as follows; air channels having two pieces of 45 × 260 cm, two pieces of 40 × 260 cm areas and 1.3 m length, one tube-body-type heat exchanger and two radial fans. One of these fans provides hot air to the evaporator section, while the other fan provides cold air to the condenser section. The velocity and mass flow rate of the air flow passing through the air channels was measured with an anemometer. As shown in Table 1, during the experimental study, the velocities and flows of both hot and cold air revealed similarities and differences.
Schematic view of the experimental setup is shown in Figure 3 while general view of the experimental setup is shown in Figure 4. As shown in Figure 4, 14 measurement points were determined in the experimental setup: one measurement point is on the evaporator inlet channel, one measurement point is on the evaporator outlet channel, one measurement point is on the condenser inlet channel, one measurement point is on the condenser outlet channel, five measurement points are on the heat pipes in the evaporator section, five measurement points are on the heat pipes in the condenser section. Temperatures were measured by K-type thermocouples with 0.1ºC of accuracy. Thermocouples were placed in the middle of the evaporator and condenser sections, and air channels. The data obtained in these measurements were recorded on the data collection (ORDEL UDL 100) device in the experimental setup.

Table 1. Air velocity and flow values used in experiments

	1. Stage velocity test	**2. Stage velocity test**	**3. Stage velocity test**	**4. Stage velocity test**	**5. Stage velocity test**	**6. Stage velocity test**	**7. Stage velocity test**
Hot Air Velocity	2.03 m/s	2.24 m/s	1.90 m/s	1.70 m/s	2.05 m/s	2.15 m/s	1.15 m/s
Hot Air Flow	111 g/s	122 g/s	103 g/s	92 g/s	112 g/s	117 g/s	61 g/s
Cold Air Velocity	2.03 m/s	1.12 m/s	1.50 m/s	1.50 m/s	1.03 m/s	0.98 m/s	0.98 m/s
Cold Air Flow	84 g/s	45 g/s	61 g/s	61 g/s	42 g/s	40 g/s	40 g/s

The working fluid used in the system was kept constant at 24 ml without considering the flow of the fluid used. This value corresponds to the 1/3 of the evaporator volume in the heat pipe. The pressure within the heat pipe was adjusted as 10 kPa while charging the heat pipe with working fluid. During experiments, the heat pipe heat exchanger bundle was held horizontally at an angle of 90º. In the evaporation section, the applied heating power was 3 kW and 6 kW. The velocities and flows of the cooling air used in the condensation section were set at five different values, as shown in Table 1. After the heat pipes in the nanofluid setup were vacuumed, each of them was filled with first distilled water and then prepared working nanofluid namely 2 % TiO_2 by mass.

4. Results and Discussion

The efficiency of the heat pipe exchanger was investigated by using distilled water and TiO_2 working fluids at different inlet temperatures (3 kW and 6 kW) in the evaporator section and different air flows in the condenser section.

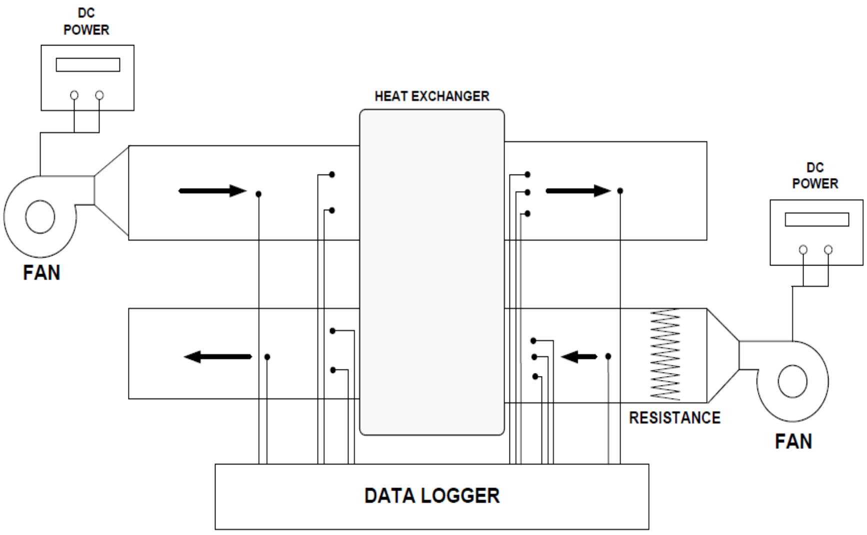

Figure 3. Schematic view of the experimental setup

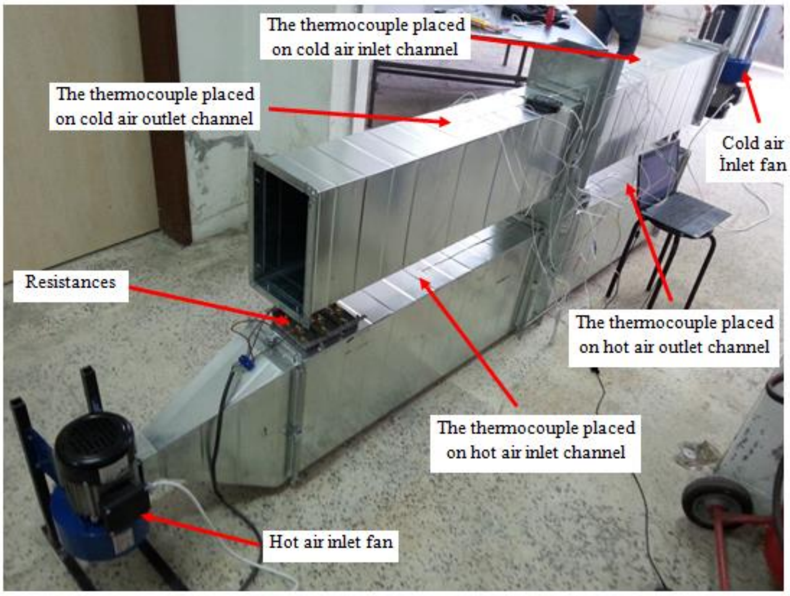

Figure 4. General view of the experimental setup

The amount of heat transferred to intake air was calculated by using inlet and outlet temperatures of the intake air, as well as mass flow rate and specific heat values of the air channel:

$$\dot{Q}_C = \dot{m}_C \, . \, \dot{c}_p \, . (T_{out} - T_{in}) \tag{1}$$

Thermal performance of a heat pipe is defined as the ratio of heat output of the condenser section to the heat input of the evaporator section and calculated as follows [14] :

$$\eta = \frac{\dot{Q}_c}{\dot{Q}_e} = \frac{\dot{Q}_{out}}{\dot{Q}_{in}} \tag{2}$$

The efficiency of a heat pipe heat exchanger is found by using the equation below [15]:

$$\varepsilon = \frac{V_{ein}\, c_{p1}\, (t_{ein} - t_{eout})}{V_{cin}\, c_{p2}\, (t_{ein} - t_{cin})} \times 100 \tag{3}$$

Figure 5a compares efficiency values of the liquids used in the experimental study for different intake (hot) air velocities and flows at the heating power of 3 kW, while Figure 5b shows the comparison of efficiency values of the liquids used in the experimental study for different outlet (cold) air velocities and flows at the heating power of 3 kW. The experimental data were obtained after minimum 30-minute runs at each velocity stage. During experiments, the highest efficiency value of working fluids was achieved for the hot air inlet velocity of 1.9 m/s and the fresh air inlet velocity of 1.5 m/s. The low hot air velocity value reduces the amount of transferred air; thereby decreasing the amount of transferred heat. The maximum efficiency values obtained were approximately 65% and 61% for TiO_2 and distilled water, respectively. Efficiency values obtained for different air velocities at 3 kW are compared in recovery charts in Figure 6a where it is observed that the maximum recovery achieved was 50% for the air velocity of 2.03 m/s. Figure 6b compares efficiency values of the liquids used in the experimental study for different intake (hot) air velocities and flows at the heating power of 6 kW, while Figure 7a shows the comparison of efficiency values of the liquids used in the study for different outlet (cold) air velocities and flows at the heating power of 6 kW. During experiments, the highest efficiency value of working fluids was obtained for the hot air inlet velocity of 1.15 m/s and the fresh air inlet velocity of 0.98 m/s. The maximum efficiency values obtained were found to be approximately 44% and 40% for TiO_2 and distilled water, respectively.

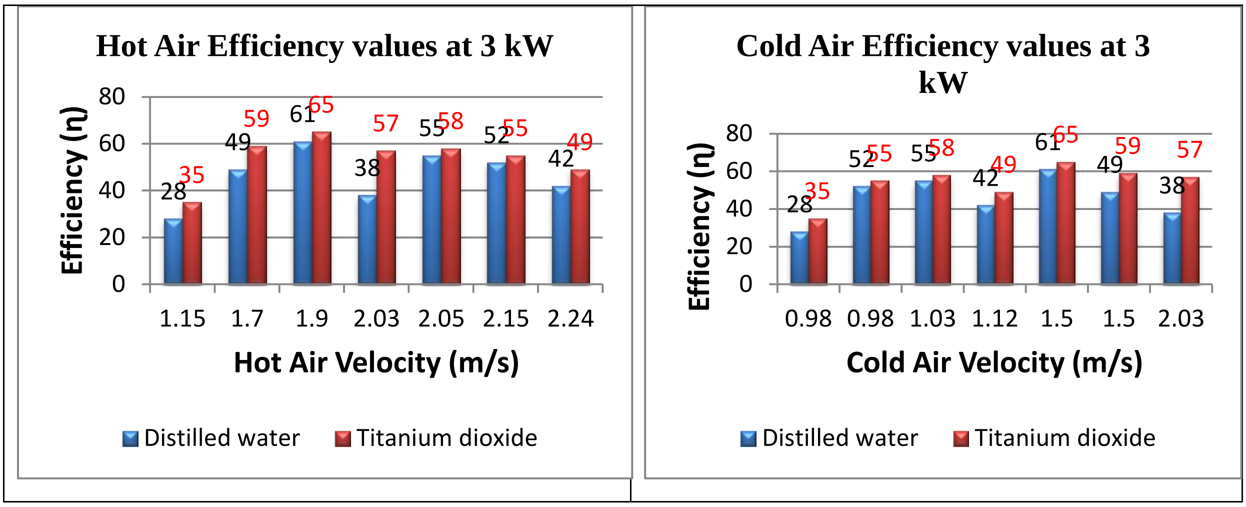

Figure 5. Comparison of efficiencies of working fluids for various hot air velocities (a) and cold air velocities (b) at 3 kW

Recovery percent is a percental value of a ratio. That is, the rate of obtained temperature increments in cold (fresh) air when using nanofluid and deionized water as the working fluid in heat pipe.

Efficiency values obtained for different air velocities at 6 kW are compared in recovery charts in Figure 6b where it is observed that the maximum recovery was 23.53% for the air velocity of 1.50 m/s. The main reason for these improvements was the presence of TiO_2 particles in the nanofluid used as the working fluid, and these nanoparticles increased the amount of heat transfer. Increased heat transfer through the use of nanofluids was also demonstrated in these experiments. It was observed that the heat transfer rate increased with increasing the heating power of working fluids, and the return velocity of the working fluid to the evaporator section also increased. It is seen that the increase in the velocity of the working liquids in the vacuum environment was adversely affected when comparing recovery percent values in Figures 6a and 6b. Experiments conducted at the heating power of 6 kW and air velocity of 0.98 m/s are the most apparent indications of this phenomenon.

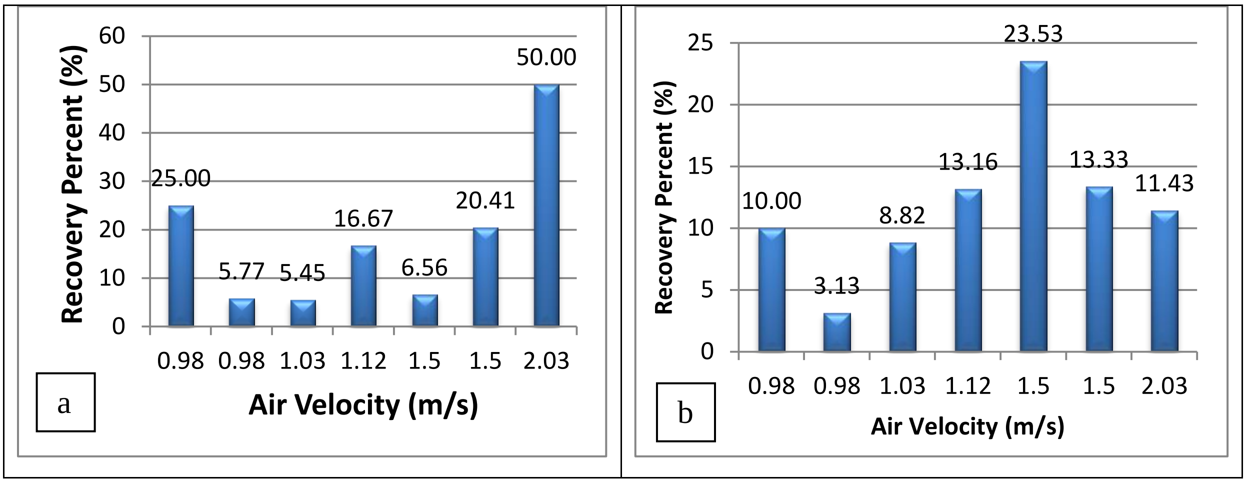

Figure 6. Recovery percent for 3 kW (a) and 6 kW (b)

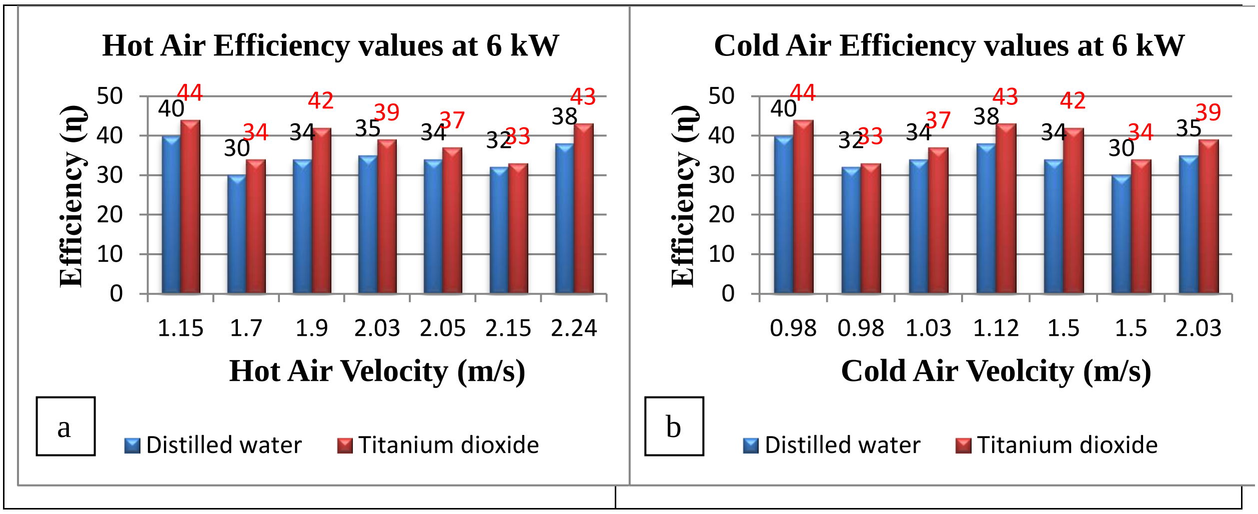

Figure 7. Comparison of efficiencies of working fluids for various hot air velocities(a) and cold air (b) at 6 kW

As a result of experiments, it was observed that the efficiency of the heat pipe increased with the use of nanofluid as the working fluid. Also, as a result of these experiments, it was observed that TiO_2 nanofluid has more stable than the others. Any coagulation and deposition in the nanofluid were not occurred for two months due to the its gelatinization. The highest efficiency (η = 65%) was achieved in the experiment using TiO_2 nanofluid at the heating power of 3 kW and air velocity of 1.5 m/s.

The thermal resistance between the evaporator and the condenser section is defined as:

$$R = \frac{\Delta T}{\dot{Q}_{in}} \tag{4}$$

Where $\dot{Q}_{in}=\dot{Q}_{e}$ is the heating power (evaporator section), ΔT is the temperature difference between mean temperature of the evaporator wall (the average of the temperatures measured at locations in the evaporator) and mean temperature of the condenser wall.

Consideration of Figure 8 yields insights about the heat-related improvement in the thermal resistance. According to Figure 8, the decrease in thermal resistance is inversely proportional to the increase in heat. Use of a nano-fluid as the working fluid yields similar relationships. However, this study in its essence is the characterization of to what degree the TiO_2 content decreases the thermal resistance in comparison to water. Figure 8 displays this quite clearly. The differences between the improvement ratios, as we have explained before, originate from the ΔT value. The differences between the improvement ratios are also confirmed by a literature-based explanation via the following paragraph;

By deploying the nanofluid as the working fluid, the thermal resistance of the heat pipe tended to decrease (Fig. 8). Why the thermal resistance of the heat pipe decreases can be explained as being a consequence of the fact that thermal resistance decrement in the heat pipe stems from the inducement of vapor bubbles at the liquid-solid interface [16]. As the bubble nucleation size increases, the thermal resistance also increases, deteriorating the heat transfer from the solid surface to the liquid [17]. The suspended nanoparticles are prone to hover over a vapor bubble during its formation. Consequently, it is expected that the nucleation size of the vapor bubble will be substantially smaller within a fluid having suspended nanoparticles than a plain fluid [18].

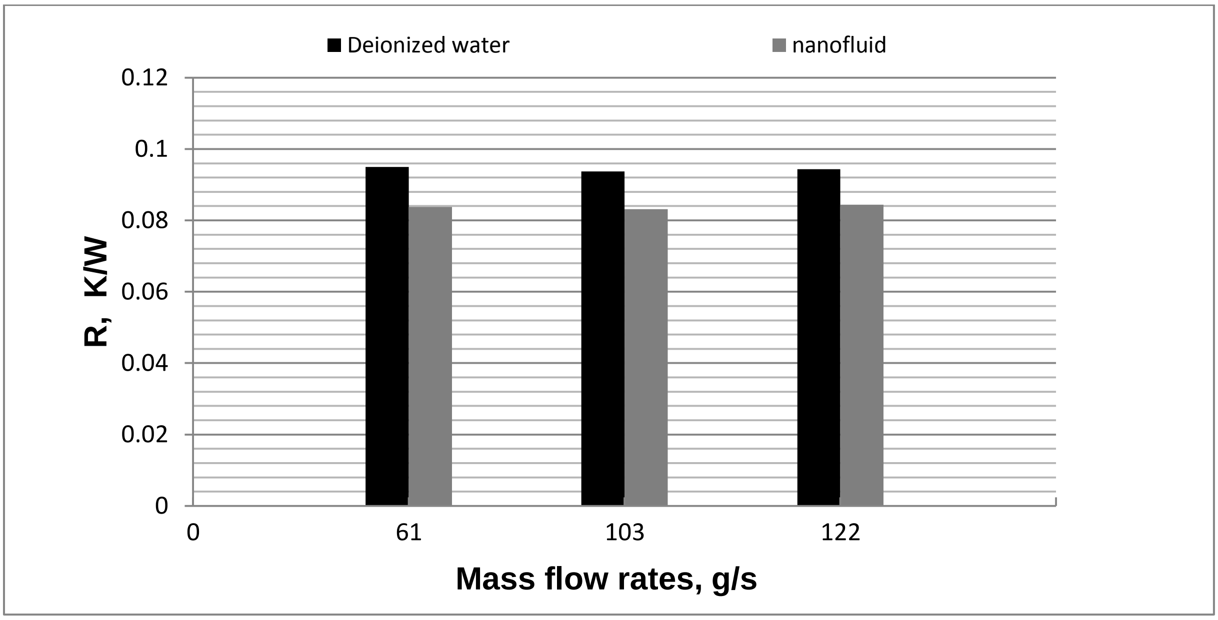

Fig.8. Thermal resistance of heat pipe

5. Conclusion

In this experimental study carried out to determine the performance of the heat pipe heat exchanger, working fluids were TiO_2 nanofluid obtained by mixing 2 % TiO_2 (m/m) particles with distilled water and distilled water. The evaluation of the performance data of the heat pipe heat exchanger is as follows:

i. Increasing air velocity of the evaporator section up to 2.0 m/s at the heating power of 3 kW increased the efficiency.
ii. Increase in the amount of heat input decreased the efficiency in heat pipe heat exchangers.
iii. Titanium dioxide was more effective than distilled water in terms of thermal and energy recovery performances.
iv. Heat pipes were manufactured as stripeless and gravity force was utilized to return the fluid.

Nomenclature

DW : Distilled water

PA : Pure acetone

HTSSHP: High temperature special shaped heat pipe

FHP : Flat-plate heat pipe

CHPs : Cylindrical heat pipes

NTU : Number of Transfer units.

R134A : Freon 134

R410A : Freon 410A

Acknowledgement

This work was supported by Scientific Research Projects Coordination Unit of Karabük University. Project Number: "KBÜ-BAP-14/2-DR-018".

References

[1] Noie-Baghban, S. H., and G. R. Majideian, Waste Heat Recovery Using Heat Pipe Heat Exchanger (Hphe) For Surgery Rooms in Hospitals. *Applied Termal Engineering 20* (2000), pp.1271-1282.

[2] Faghri, A, *Heat pipe science and technology,* Global Digital Press, 1995

[3] Faghri, A, Heat Pipes: Review, Opportunities and Challenges. *Frontiers in Heat Pipes 5* (2014) pp.1-48.

[4] Tosun, H., E. Söylemez, and M. Özşen. 2013. Isı Borulu Isı Geri Kazanım ve Nem Alma Ünitesi. *11. Ulusal Tesisat Mühendisliği Kongresi*, *1* (2013), pp.373-391. (in Turkish)

[5] Zhu, Y., X. Cui, H. Han, and S. Sun., The study on the difference of the start-up and heat-transfer performance of the pulsating heat pipe with water–acetone mixtures. *International Journal of Heat and Mass Transfer 77* (2014), pp. 834-842.

[6] Riffat, S. B., and G. Gan., Determination of effectiveness of heat-pipe heat recovery for naturally-ventilated buildings. *Applied Thermal Engineering 18* (1998), pp. 121-130.

[7] Wang, X., T. Ma, Y. Zhu, H. Chen, and J. Zeng., Experimental investigation on startup and thermal performance of a high temperature special-shaped heat pipe coupling the flat plate heat pipe and cylindrical heat pipes. *Experimental Thermal and Fluid Science 77* (2016), pp. 1-9.

[8] Zhu, Z., H. Fan, and C. Zhang. 2016. Experimental investigations on the effectiveness of micro heat pipe array heat exchanger for heat recovery for residential building. *Applied Thermal Engineering 102* (2016), pp. 980-988.

[9] Sadeghinezhad, E. et al., Experimental investigation of the effect of graphene nanofluids on heat pipe thermal performance. *Applied Thermal Engineering 100* (2016), pp.775-787.

[10] Noie, S. H., Investigation of thermal performance of an air-to-air thermosyphon heat exchanger using epsilon-NTU method. *Applied Thermal Engineering 26*: (2016), pp.559-567.

[11] Orr, B., B. Singh, L. Tan, and A. Akbarzadeh., Electricity generation from an exhaust heat recovery system utilising thermoelectric cells and heat pipes. *Applied Thermal Engineering 73* (2014), pp. 588-597.

[12] Gedik, E., M. Yılmaz, and H. Kurt, Experimental investigation on the thermal performance of heat recovery system with gravity assisted heat pipe charged with R134a and R410A. *Applied Thermal Engineering 99* (2016), pp. 334-342.

[13] ALARKO, Enerji Geri Kazanim Havalandirmasi. *ALARKO Carrier Teknik Bülten 2013, 55.(in Turkish)*

[14] Sözen, A. et al., A comparative investigation on the effect of fly-ash and alumina nanofluids on the thermal performance of two-phase closed thermo-syphon heat pipes. *Applied Thermal Engineering 96* (2016), pp. 330-337.

[15] Zhang, J. et al., Experimental study on the heat recovery characteristics of a new-type flat micro-heat pipe array heat exchanger using nanofluid. *Energy Conversion and Management 75* (2013), pp. 609-616.

[16] G. Huminic, A. Huminic, Heat transfer characteristics of a two-phase closed thermosyphons using nanofluids, *Exp. Therm. Fluid Sci. 35* (2011), pp. 550–557.

[17] J.G. Coiller, J.R. Thome, Convective Boiling and Condensation, Clarendon Press, Oxford, U.K., 1996.

[18] S.H. Noie, S.Z. Heris, M. Kahani, S.M. Nowee, Heat transfer enhancement using Al2O3/water nanofluid in a two-phase closed thermosyphon, *Int. J. Heat Fluid Flow 30* (2009), pp. 700–705.

Submitted: 3.07.2017.
Revised:29.11.2017.
Accepted: 4.12.2017.