$$\dot{E}x_{cond} = \dot{m}_a c_{p,a} \begin{bmatrix} (T_{out} - T_{in}) \\ -T_0 \ln \frac{T_{out}}{T_{in}} \end{bmatrix} \quad (12)$$

The exergy $(\dot{E}x_{heater})$ supplied by the electricity to the evaporator can be calculated as follow:

$$\dot{E}x_{heater} = \dot{Q}_{heater} \quad (13)$$

The exergy efficiency (η_{II}) is is defined as the ratio of useful exergy output rate to the total exergy input rate.

$$\eta_{II} = \frac{\text{Useful exergy output rate}}{\text{Supplied exergy rate}} = \frac{\dot{E}x_{cond}}{\dot{E}x_{heater}} \quad (14)$$

4 Uncertainty analysis

In order to calculate the uncertainty of any parameter such as energy and exergy performance, the uncertainty interval for independent experimental measured variables (xi), should be estimated by following equation [24]:

$$e_{\eta i} = \frac{Xi}{\eta} \frac{\partial \eta}{\partial xi} e_{\eta i} \quad (15)$$

where η is calculated parameter by using measurable quantities, exi is measurement error for experimental measured variable and should be determined by dividing the measurement accuracy to the minimum measured value and eηi is the approximate possible error introduced in calculating one parameter.

Uncertainty of the experimental data may have resulted from measuring errors of parameters such as inlet and outlet temperature and pipe diameter of the coolant air in the condenser, airflow velocity and electricity consumption are shown in Table 1.

Due to the combined effect of uncertainty intervals in all values of xi (i=1, 2,..., n) the uncertainty in a parameter such as η can be calculated by Eq. [24]:

$$e_\eta = \pm \left[\left(\left(\frac{X1}{\eta} \frac{\partial \eta}{\partial x1} e_{x1} \right)^2 + \left(\left(\frac{X2}{\eta} \frac{\partial \eta}{\partial x2} e_{x2} \right)^2 + \cdots + \left(\left(\frac{Xn}{\eta} \frac{\partial \eta}{\partial xn} e_{xn} \right)^2 \right]^{1/2} \quad (16)$$

5 Results and discussion

In literature, there are several available experimental and theoretical studies on the determination of heat transfer characteristics of various THP systems. However, in these studies, the using of Ejector in THPs have not been examined yet. Therefore, this study aims to determine experimentally the effects of ejector on energy and exergy performance in THP systems.

In experiments, measured temperature difference alterations of ETHP-SP, CTHP and ETHP-TP in relation with time for different airflow velocities, 3.3, 5.3 and 6.3 ms-1, are illustrated in Figures 3, 4 and 5 respectively.

As seen in Figure 3, 4 and 5, when airflow velocity is 3.3 ms-1, the steady state condition of the THP systems occurs after 90 min, while the steady state condition of the THP systems occurs after 60 min, when airflow velocities are 5.3 and 6.3 ms-1.

As understood from the figure 3, 4 and 5, under the steady state condition, the air outlet and inlet temperature differences for the ETHP-SP are higher than those of the CTHP and ETHP-TP at the same heat input and airflow velocity.

The energy performance is defined as the ratio of the heat transferred by the condenser to that the supplied at the evaporator. The energy performance alterations of THPs in relation with time are given in Figure 6, 7 and 8. In terms of the exergy, the performance is defined as the ratio of supplied exergy to the useful product exergy by the system and the exergy performance alterations of THPs in relation with time are presented in Figure 9, 10 and 11.

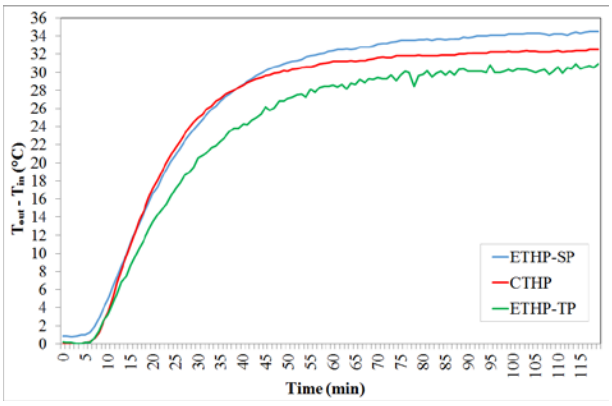

Fig. 3. Measured temperature difference values of the THPs when $V_{air} = 3.3\ ms^{-1}$

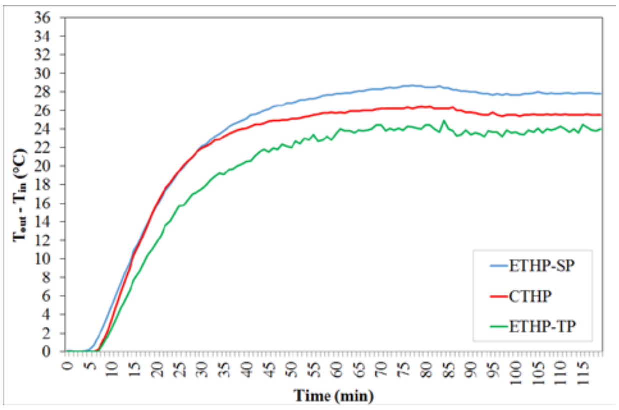

Fig. 4. Measured temperature difference values of the THPs for $V_{air} = 5.3$ ms^{-1}

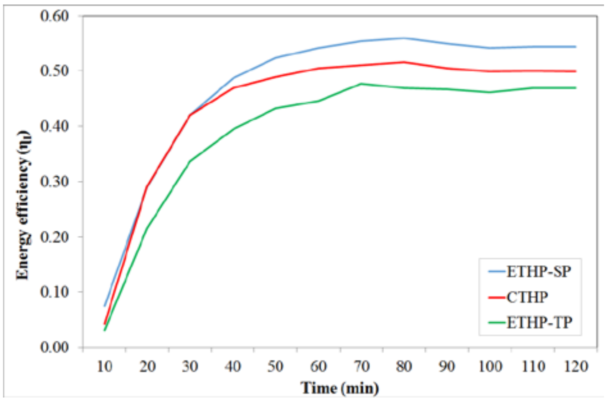

Fig. 7. Energy efficiency values of the THPs for $V_{air} = 5.3$ ms^{-1}

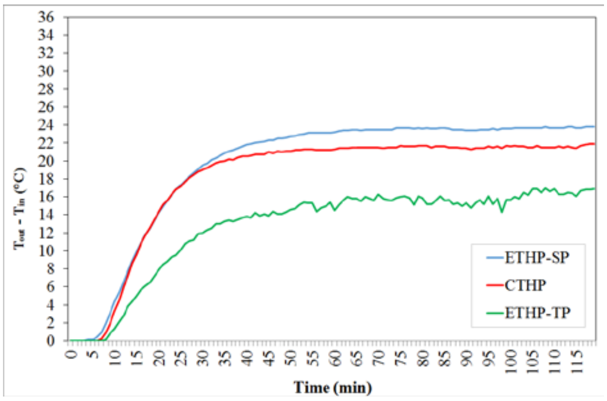

Fig. 5. Measured temperature difference values of the THPs for $V_{air} = 6.3$ ms^{-1}

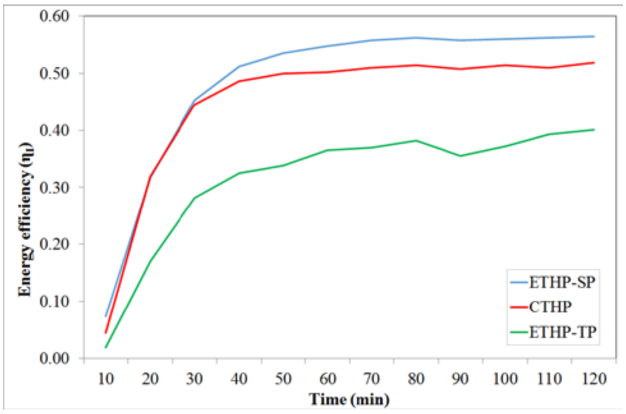

Fig. 8. Energy efficiency values of the THPs for $V_{air} = 6.3$ ms^{-1}

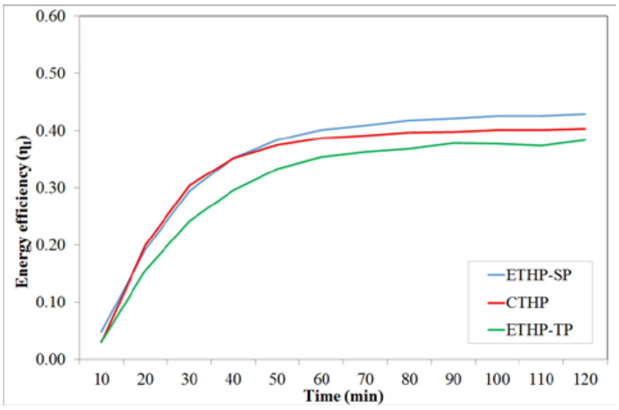

Fig. 6. Energy efficiency values of the THPs for $V_{air} = 3.3$ ms^{-1}

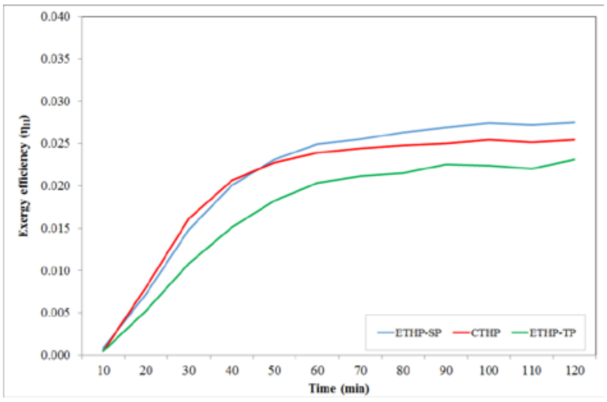

Fig. 9. Exergy efficiency values of the THPs for $V_{air} = 3.3$ ms^{-1}

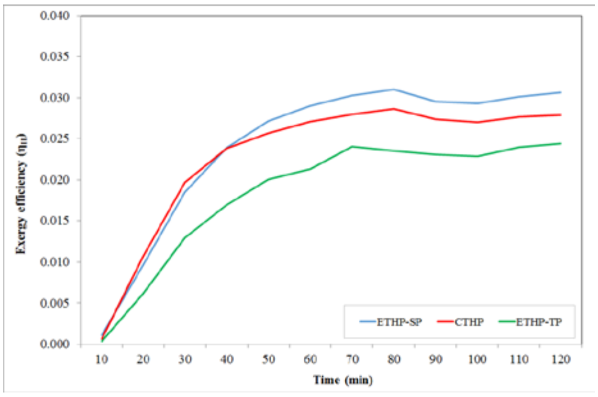

Fig. 10. Exergy efficiency values of the THPs for $V_{air} = 5.3\ ms^{-1}$

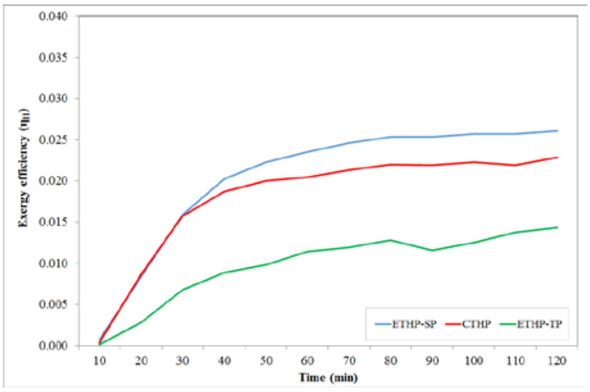

Fig. 11. Exergy efficiency values of the THPs for $V_{air} = 6.3\ ms^{-1}$

Mean energy and exergy performance of the THPs are calculated by mean values of data obtained following the beginning of the steady state conditions and these calculated values of mean energy and exergy performance are given in Table 1.

The mean energy performance of ETHP-SP, CTHP and ETHP-TP for airflow velocity which is 3.3 ms-1 were determined as 0.425, 0.401 and 0.376 respectively. Also, the mean energy performance of ETHP-SP, CTHP and ETHP-TP for airflow velocity which is 5.3 ms-1 were determined as 0.549, 0.505 and 0.467 respectively while the mean energy performance of ETHP-SP, CTHP and ETHP-TP for airflow velocity, 6.3 ms-1, were determined as 0.559, 0.510 and 0.376 respectively.

The mean exergy performance of ETHP-SP, CTHP and ETHP-TP for airflow velocity which is 3.3 ms-1 were determined as 0.027, 0.0,025 and 0.0,022 respectively. Also, the mean exergy performance of ETHP-SP, CTHP and ETHP-TP for airflow velocity, 5.3 ms-1, were determined as 0.030, 0.028 and 0.023 respectively, whereas, the mean exergy performance of ETHP-SP, CTHP and ETHP-TP for airflow velocity which is 6.3 ms-1 were determined as 0.025, 0.022 and 0.0,13 respectively.

As seen in the Table 1, under all these airflow velocities, while the lowest energy and exergy performance occur in the ETHP-TP and the highest energy and exergy performance in the ETHP-SP. As a result, under different airflow velocities which are determined 3.3, 5.3 and 6.3 ms-1, by using ejector with short vapor pipe in the conventional thermosyphon heat pipe, energy performance of CTHP increases by 5.9%, 8.7% and 9.6% respectively. In addition, exergy performance of CTHP goes up by 8%, 7.1% and 13.6% respectively.

Maximum uncertainty for mean energy and exergy performance of THPs when airflow velocity of 3.3 ms-1 is calculated. Maximum uncertainty of parameters is given in Table 2 and its results are given in Table 3. Therefore, the maximum uncertainties in energy and exergy performance of THPs are ± 0.064 and ± 0.065 respectively. Minimum uncertainties for mean energy and exergy performance of THPs when airflow velocity of 6.3 ms-1 are calculated and its values are ± 0.042 and ± 0.044 respectively.

6 Conclusions

There are many studies aiming to improve the theoretical and experimental heat transfer characteristics of different THP systems. However, neither experimental nor theoretical studies on effects of ejector on both thermosyphon heat pipes and heat pipes in general exist in literature.

The effects of ejector on the energy and exergy performance of a THP under the same heat input and filling rate conditions was investigated experimentally in this paper in the airflow velocities which were determined as 3.3, 5.3 and 6.3 ms-1.

Considering the results of the analyses, the following main conclusions can be drawn from the present study:

Table 1. The results of the mean energy and exergy performance of THPs

Airflow rate (ms^{-1})	ETHP-SP			CTHP			ETHP-TP		
	ΔT (K)	η_I	η_{II}	ΔT (K)	η_I	η_{II}	ΔT (K)	η_I	η_{II}
3.3	34.20	0.425	0.027	32.27	0.401	0.025	30.29	0.376	0.022
5.3	28.07	0.549	0.030	25.82	0.505	0.028	23.88	0.467	0.023
6.3	23.60	0.559	0.025	21.54	0.510	0.022	15.87	0.376	0.013

Table 2. Maximum uncertainty of parameters

Temperature difference	$e_{(T_{out}-T_{in})} = e_{\Delta T} = \pm \frac{0.25}{34.2} = \pm 0.007$
Airflow velocity	$e_v = \pm \frac{0.2}{3.3} = \pm 0.06$
Air inlet and outlet pipe diameter	$e_d = \pm \frac{0.05}{17} = \pm 0.003$
Electricity consumption	$e_{e,c} = \pm \frac{0.1}{60.6} = \pm 0.002$

Table 3. Maximum uncertainty of calculated parameters

$$e_\rho = \pm[(e_{\Delta T})^2]^{0.5} = \pm 0.007 = \pm 0.7\ \%$$

$$e_A = \pm[4(e_d)^2]^{0.5} = \pm 0.012 = \pm 1.2\ \%$$

$$e_m = \pm[(e_{\Delta T})^2 + (e_A)^2 + (e_v)^2]^{0.5} = \pm 0.06 = \pm 6\ \%$$

$$e_{c,p} = \pm[(e_{\Delta T})^2]^{0.5} = \pm 0.007 = \pm 0.7\ \%$$

$$e_{Qcond} = \pm\left[(e_m)^2 + (e_{c,p})^2 + (e_{\Delta T})^2\right]^{0.5} = \pm 0.06 = \pm 6\ \%$$

$$e_{Qheater} = \pm\left[(e_{e,c})^2\right]^{0.5} = \pm 0.002 = \pm 0.2\ \%$$

$$e_{\eta_I} = \pm\left[(e_{Qcond})^2 + (e_{Qheater})^2\right]^{0.5} = \pm 0.064 = \pm 6.4\ \%$$

$$e_{Ex,cond} = \pm\left[(e_m)^2 + (e_{c,p})^2 + (e_{\Delta T})^2 + (e_{\Delta T})^2\right]^{0.5} = \pm 0.06 = \pm 6\ \%$$

$$e_{Ex,heater} = \pm\left[(e_{e,c})^2\right]^{0.5} = \pm 0.002 = \pm 0.2\ \%$$

$$e_{\eta_{II}} = \pm\left[(e_{Ex,cond})^2 + (e_{Ex,heater})^2\right]^{0.5} = \pm 0.064 = \pm 6.4\ \%$$

- Maximum energy performance of ETHP-SP, CTHP and ETHP-TP come out as 0.559, 0.510 and 0.376 respectively when the airflow velocity is 6.3 ms-1.
- Maximum exergy performance of ETHP-SP, CTHP and ETHP-TP occur as 0.030, 0.028 and 0.023 respectively when the airflow velocity is 5.3 ms-1.
- Of all airflow velocities, the ETHP-SP has the highest energy and exergy performance while ETHP-TP has the lowest energy and exergy performance.
- It is found that the use of ejector in CTHP systems increases the mean energy performance by 9.6% and mean exergy performance by 13.6%.

The use of ejector in thermosyphon heat pipes is proved to be effective in this study. For future analysis the use of ejector in thermosyphon heat pipes with different filling rates, working fluids, heat input, aspect ratio, inclination angle, pipe lengths and materials can improve the efficiency of these systems. And even the place the ejector is located can make difference. Also, the diameters of ejector vapor pipe and ejector liquid pipe can lead to an increase in energy and exergy performance.

References

[1] P. Terdtoon, N. Waowaew, P. Tantakom, Internal flow patterns of an inclined, closed two-phase thermosyphon at critical state: case study 1, effect of aspect ratio, *Experimental Heat Transfer*, Vol.12, No.4, 1999, pp. 347–358.

[2] P. Terdtoon, N. Waowaew, P. Tantakom, Internal flow patterns of an inclined, closed two-phase thermosyphon at critical state: case study 2, effect of Bond number, *Experimental Heat Transfer*, Vol.12, No.4, 1999, pp. 359–373.

[3] K.S. Ong, Md. Haider-E-Alahi, Performance of a R-134a-Filled Thermosyphon, *Applied Thermal Engineering*, Vol.23, No.18, 2003, pp. 2373–2381.

[4] H.M.S. Hussein, H.H. El-Ghetany, S.A. Nada, Performance of wickless heat pipe flat plate solar collectors having different pipes cross sections geometries and filling ratios, *Energy Conversion and Management*, Vol.47, No.11-12, 2006, pp. 1539–1549.

[5] B. Jiao, L.M. Qiu, X.B. Zhang, Y. Zhang, Investigation on the effect of filling ratio on the steady-state heat transfer performance of a vertical two-phase closed thermosyphon, *Applied Thermal Engineering*, Vol.28, No.11-12, 2008, pp. 1417–1426.

[6] S. Khandekar, Y.M. Joshi, B. Mehta, Thermal Performance of Closed Two-Phase Thermosyphon Using Nanofluids, *International Journal of Thermal Sciences*, Vol.47, No.6, 2008, pp. 659–667.

[7] S.H. Noie, S.Z. Heris, M. Kahani, S.M. Nowee, Heat Transfer Enhancement Using Al2O3/Water Nanofluid In A Two-Phase Closed Thermosyphon, *International Journal of Heat and Fluid Flow,* Vol.30, No.4, 2009, pp. 700–705.

[8] G. Huminic, A. Huminic, I. Morjan, F. Dumitrache, Experimental Study of The Thermal Performance of Thermosyphon Heat Pipe Using Iron Oxide Nanoparticles, *International Journal of Heat and Mass Transfer,* Vol.54, No.1-3, 2011, pp. 656–661.

[9] G. Huminic, A. Huminic, Numerical Study On Heat Transfer Characteristics of Thermosyphon Heat Pipes Using Nanofluids, *Energy Conversion and Management,* Vol.76, 2013, pp. 393–399.

[10] A. Kamyar, K.S. Ong, R. Saidur, Effects of Nanofluids On Heat Transfer Characteristics of A Two-Phase Closed Thermosyphon, *International Journal of Heat and Mass Transfer***,** Vol.65, 2013, pp. 610–618.

[11] M.H. Buschmann, U. Franzke, Improvement of Thermosyphon Performance By Employing Nanofluid, *International Journal of Refrigeration,* Vol.40, 2014, pp. 416–428.

[12] S.Z. Heris, F. Mohammadpur, A. Shakouri, Effect of Electric Field on Thermal Performance of Thermosyphon Heat Pipes Using Nanofluids, *Materials Research Bulletin,* Vol.53, 2014, pp. 21–27.

[13] Z.Q. Long, P. Zhang, Heat Transfer Characteristics of Thermosyphon With N2–Ar Binary Mixture Working Fluid, *International Journal of Heat and Mass Transfer***,** Vol.63, 2013, pp. 204–215.

[14] Q. Baojin, Z. Li, X. Hong, S. Yan, Heat Transfer Characteristics of Titanium/Water Two-Phase Closed Thermosyphon, *Energy Conversion and Management*, Vol.50, No.9, 2009, pp. 2174–2179.

[15] Y.S. Lee, Y.P Lee, Y. Lee. The Effects of Surface Tension and Wire Diameter on The Rise Velocity of .a Bubble in a Miniature Two-Phase Closed, Thermosyphon, *Applied Thermal Engineering*, Vol.16, 1996, pp. 655–668.

[16] L. Lin, A. Faghri, Steady-State Performance in a Thermosyphon with Tube Separator, *Applied Thermal Engineering*, Vol.17, 1997, pp. 661–679.

[17] L. Lin, A. Faghri, An Analysis of Two-Phase Flow Stability in a Thermosyphon with Tube Separator, *Applied Thermal Engineering*, Vol.18, 1998, pp. 441–455.

[18] K.A. Joudi, A.M. Witwit, Improvements of Gravity Assisted Wickless Heat Pipes. *Energy Conversion and Management*, Vol.41, 2000, pp. 2041-2061.

[19] M. Zhang, Z. Liu, G. Ma, The Experimental Investigation on Thermal Performance of a Flat Two-Phase Thermosyphon, *International Journal of Thermal Sciences*, Vol.47, 2008 pp. 1195–1203.

[20] S. Rittidech, W. Srimuang, Correlation to Predict Heat-Transfer Characteristics of a Vertical Flat Thermosyphon (VFT) At Normal Operating Conditions, *International Journal of Heat and Mass Transfer*, Vol.53, 2010, pp. 5984–5987.

[21] H. Peng, X. Ling, J. Li, Numerical Simulation and Experimental Verification on Thermal Performance of a Novel fin-Plate Thermosyphon, *Applied Thermal Engineering*, Vol.40, 2012, pp. 181-188.

[22] K.S. Ong, Thermal performance of solar air heaters-experimental correlation, *Solar Energy*. Vol.55, No.3, 1995, pp. 209–220.

[23] I. Dincer, M.A. Rosen, *Exergy: energy environment and sustainable development*, Oxford: Elsevier, 2007.

[24] J.D. Holman, *Experimental Methods for Engineers*, New York: McGraw-Hill, 1989.