Thermal Performance Comparison of Ejecting Thermosyphon Heat Pipe with Conventional Thermosyphon Heat Pipe

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Abstract: This study, by using ejector, deals with improvement of energy and exergy performance of thermosyphon heat pipes. For this aim, an ejecting thermosyphon heat pipe with short vapor pipe (ETHP-SP), an ejecting thermosyphon heat pipe with tall vapor pipe (ETHP-TP) and a conventional thermosyphon heat pipe (CTHP) are designated and manufactured for heating air and they are experimentally investigated under different airflow velocities but with the same heat input and filling rate. While the lowest energy and exergy performance occur in the ETHP-TP, the highest energy and exergy performance occur in the ETHP-SP. Under different airflow velocities which are determined as 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.

Key-Words: thermosyphon heat pipe- ejector- energy- exergy- performance- uncertainty

1 Introduction

Thermosyphon Heat Pipe (THP), a gravity-assisted wickless heat pipe, utilizes the evaporation and condensation of the working fluid inside to transport heat and is called as two-phase close thermosyphon (TPCT). In contrast to the conventional heat pipe using capillary force to return the liquid to evaporator, a THP uses gravitation to return the condensate.

It has simpler structure, smaller thermal resistance, higher efficiency and lower manufacturing cost. Also, it contains no mechanical moving parts and typically requires no maintenance. Owing to these advantages, the THP has been widely used in many fields, such as industrial heat recovery, electronic component and turbine blade cooling, solar energy systems, climatization processes, preservation of permafrost, deicing of roadways, and so on.

The performance of a THP is significantly affected by structure and geometry, inclination angle, vapor temperature and pressure, filling ratio and thermo physical properties of working fluid.

There are many studies on the heat transfer analyses of the THP system in the literature. In order to improve performance of THP systems, investigators have focused on working fluids, filling ratio, inclination angle and components of the THP systems.

There are many studies in the literature dealing with theoretical and experimental heat transfer characteristics of various THP systems. This study is different from the previously conducted ones as follows: (i) the study using ejector to improve the performance of thermosyphon heat pipe (ETHP), (ii) the study using two types of ejector in THP systems, ejecting thermosyphon heat pipe with short vapor pipe (ETHP-SP) and ejecting thermosyphon heat pipe with tall vapor pipe (ETHP-TP), (iii) the comparison of conventional thermosyphon heat pipe (CTHP) with ejecting thermosyphon heat pipe with short vapor pipe (ETHP-SP) and ejecting thermosyphon heat pipe with tall vapor pipe (ETHP-TP), (iv) the study comparing the energy and exergy performance analysis of THPs mentioned above (ETHP-SP, ETHP-TP and CTHP).

2 Experimental apparatus

In order to improve the performance of THP systems, ejecting thermosyphon heat pipe with short vapor pipe (ETHP-SP), ejecting thermosyphon heat pipe with tall vapor pipe (ETHP-TP) and conventional thermosyphon heat pipe (CTHP) were designated and manufactured. Also, this paper compared the energy and exergy performance results of CTHP with ETHP-SP and ETHP-TP.

The schematic view and a photograph of the THP systems investigated in this study was illustrated in Fig. 1 and Fig. 2 respectively.

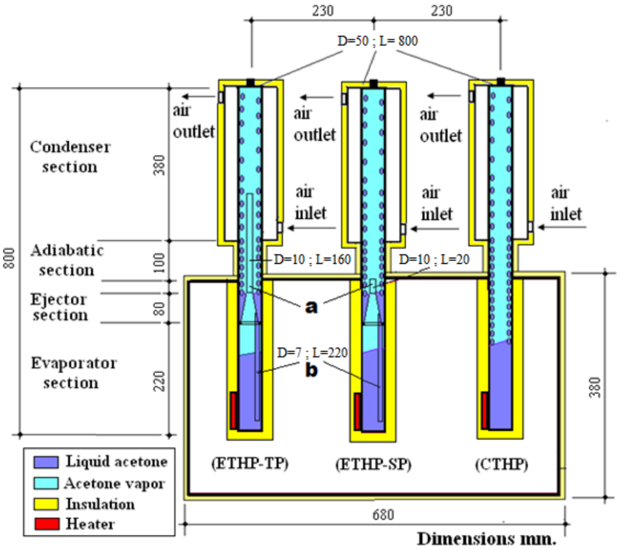

Fig. 1. The schematic view of the THP systems

Fig 2. The photograph of the THP systems

The THPs are made with the 800 mm long, Ø 50 mm aluminum pipe. The evaporator section of each of them was electrically heated by cartridge heaters which have the same power. The 380 mm long condenser section consisted of an air-cooled jacket surrounding the pipe. The jacket was an Ø 70 mm PVC pipe with Ø 25 mm PPRC pipe inlet and outlet connections located diagonally across each other to induce swirl at the inlet. After evacuating the inside of the THPs, volume of 30 ml of the acetone was charged evenly into the each of THP. In order to minimize the heat losses, the evaporator section, adiabatic section and condenser of the THPs were insulated by rock wool having a thickness of 20 mm.

In the experiments, electricity consumption, airflow velocity, the air inlet and outlet temperatures of the coolant air and ambient temperature are measured. Coolant air was supplied via a fan device. Experiments were performed using three different airflow velocities, 3.3, 5.3 and 6.3 ms^{-1}.

The measurement of temperatures is carried out by K-type thermocouples (error ±0.25 °C). Also, so as to measure coolant airflow velocities, an anemometer (error ± 0.2 ms^{-1}) is employed. Furthermore, to measure electricity consumption, electronic electricity meter (error ± 0.1 kWh) is put into use. The measured data are recorded in every one minute in the course of two hours.

3 Thermodynamic analyses of THPs

3.1 Energy analysis of THPs

In the conventional thermosyphon heat pipes, working fluid (acetone) in fluid phase in the evaporator vaporizes via heat input ($\dot{Q}_{heater}$) supplied by the electrical resistance and acetone vapor rises through the evaporator and reaches to condenser. Here, it conducts its latent heat to air and condensates in the top section of condenser and turns back into evaporator.

As for the ejecting thermosyphon heat pipes designated and manufactured by us, working fluid (acetone) in fluid phase in the evaporator vaporizes via heat input ($\dot{Q}_{heater}$) supplied by the electrical resistance and acetone vapor rises through the evaporator and reaches to condenser after passing ejector and ejector vapor pipe (a) which enable the velocity of acetone vapor to increase and to reach the condenser faster. Latent heat of acetone vapor is conducted to air and it condensates in the top section of condenser and turns back into evaporator passing ejector liquid pipe (b) which prevents the crash of liquid acetone with acetone vapor.

The heat transferred from condenser in the thermosyphon heat pipe is taken equal to the heat transferred to the cooling air at steady state conditions. This is determined by measuring the temperature difference and the airflow velocity of the cooling air and then, applying the following simple equation;

$$\dot{Q}_{cond} = \rho_a V_a A_c C_{p,a}(T_{out} - T_{in}) \quad (1)$$

where V_a is the airflow velocity (ms^{-1}), A_c is the cross sectional area (m^2), T_{out} is the outlet temperature of cooling air (K), T_{in} is the inlet temperature of cooling air (K). ρ_a is the density of air ($kg\ m^{-3}$) and $C_{p,a}$ is the specific heat of air ($kJ\ kg^{-1}\ K^{-1}$) and it can be calculated from equations given by Ong according to mean fluid temperature ($T_m = (T_{in} + T_{out})/2$) respectively [22].

$$\rho_a = 1.1774 - 0.00359(T_m - 300) \quad (2)$$

$$C_{p,a} = 1.0057 + 0.000066(T_m - 300) \quad (3)$$

Heat losses to environment from the evaporator are neglected due to the insulation of evaporator. Therefore, the heat input to the evaporator supplied by the electricity is calculated as follows:

$$\dot{Q}_{heater} = \frac{\dot{W}_{elect}}{\eta_{elect}} \quad (4)$$

In Eq. (4), $\dot{W}_{elect}$ is electrical consumption [kW] and η_{elect} is electricity efficiency and it is assumed as 0.99.

The energy efficiency (η_I) is defined as the ratio of the heat transferred by the condenser to that supplied at the evaporator:

$$\eta_I = \frac{\dot{Q}_{cond}}{\dot{Q}_{heater}} \quad (5)$$

3.2 Exergy analysis of THPs

Energy analysis method alone is not enough to understand of all the aspects of energy utilization processes of the systems. Also, it does not quantify the usefulness or quality of the various energy streams flowing through a system and exiting as products and wastes. So, exergy analysis method, which bases on the first and second laws thermodynamics, is used to understand and improve the real efficiencies of the system. Exergy is also defined as potential or quality of energy. With exergy analysis, it is possible to make sustainable quality assessment of energy for any thermodynamic system. Furthermore, it can be said that the major aims of exergy analysis are to identify real efficiencies (also known as exergy efficiency) and to determine true magnitudes of exergy losses and destructions. The exergy destruction mentioned here is proportional to the entropy generation. In real process, some of exergy is destroyed due to second law of thermodynamics [23].

A general exergy balance can be given as follows:

$$\sum \dot{E}x_{in} - \sum \dot{E}x_{out} = \sum \dot{E}x_{dest} \quad (6)$$

where "$\sum \dot{E}x_{in}$", "$\sum \dot{E}x_{out}$" and "$\sum \dot{E}x_{dest}$" are total exergy input, total exergy output and exergy destruction rates, respectively.

The general flow exergy rate ψ can be determined by

$$\psi = \dot{m}_a\,[(h - h_0) - T_0(s - s_0)] \quad (7)$$

In Eq. (7), ''$\dot{m}_a$'' is the mass flow rate of the fluid, ''h'' is the enthalpy of the fluid in given condition, ''h_0'' is the enthalpy of the fluid in dead state condition, ''s'' is the entropy of the fluid in given condition, ''s_0'' is the entropy of the fluid in dead state condition, ''T_0'' is the dead state (reference) and it is assumed that 298 K.

The exergy change of a fluid stream can be written as:

$$\begin{aligned}\Delta\Psi &= \dot{m}_a\,(\Psi_{out} - \Psi_{in}) \\ &= \dot{m}_a\,[(h_{out} - h_{in}) - T_0(s_{out} - s_{in})] \quad (8)\end{aligned}$$

The changes in the enthalpy and the entropy of the air are expressed by

$$h_{out} - h_{in} = c_{p,a}(T_{in} - T_{out}) \quad (9)$$

$$s_{out} - s_{in} = c_{p,a}\ln\frac{T_{out}}{T_{in}} - R_a\ln\frac{P_{out}}{P_{in}} \quad (10)$$

Substituting Eqs. (9) and (10) into Eq. (8), the following equation for exergy change in condenser can be derived

$$\dot{E}x_{cond} = \dot{m}_a \begin{bmatrix} c_{p,a}(T_{in} - T_{out}) \\ -T_0(c_{p,a}\ln\frac{T_{in}}{T_{out}} \\ -R_a\ln\frac{P_{in}}{P_{out}}) \end{bmatrix} \quad (11)$$

In Eq. (11), "R_a'' is the specific ideal gas constant of the air, "P_{in}" and "P_{out}" are inlet pressure and outlet pressure for air. Eq. (11) is rearranged due to inlet and outlet pressure of air that equals to atmosphere pressure (P_{in}=P_{out}=P_{atm}),

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

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]:

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

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.

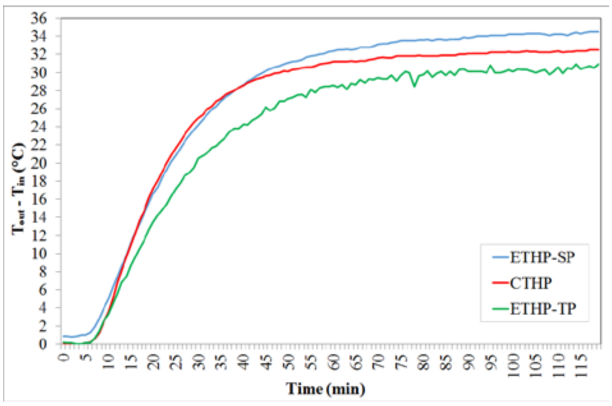

Fig. 3. Measured temperature difference values of the THPs when $V_{air} = 3.3\ ms^{-1}$

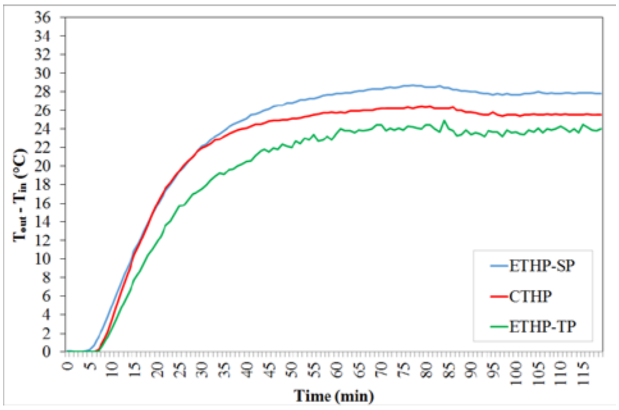

Fig. 4. Measured temperature difference values of the THPs for $V_{air} = 5.3\ ms^{-1}$

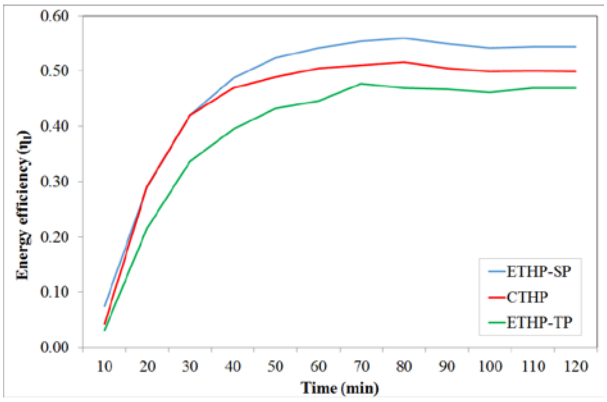

Fig. 7. Energy efficiency values of the THPs for $V_{air} = 5.3\ ms^{-1}$

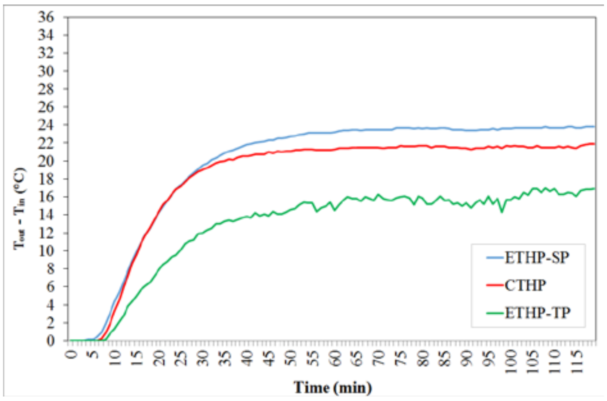

Fig. 5. Measured temperature difference values of the THPs for $V_{air} = 6.3\ ms^{-1}$

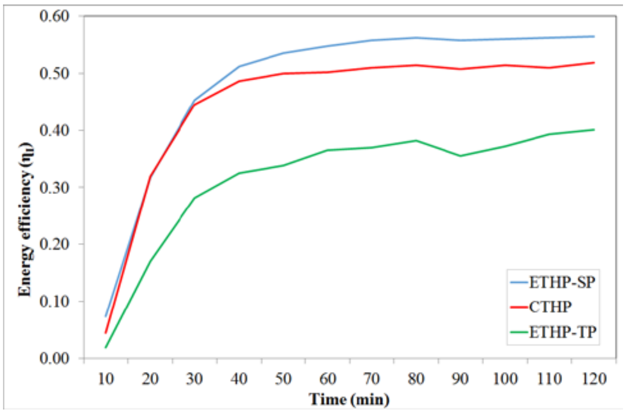

Fig. 8. Energy efficiency values of the THPs for $V_{air} = 6.3\ ms^{-1}$

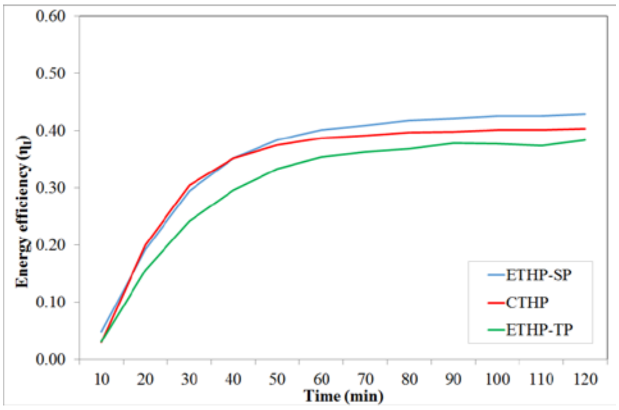

Fig. 6. Energy efficiency values of the THPs for $V_{air} = 3.3\ ms^{-1}$

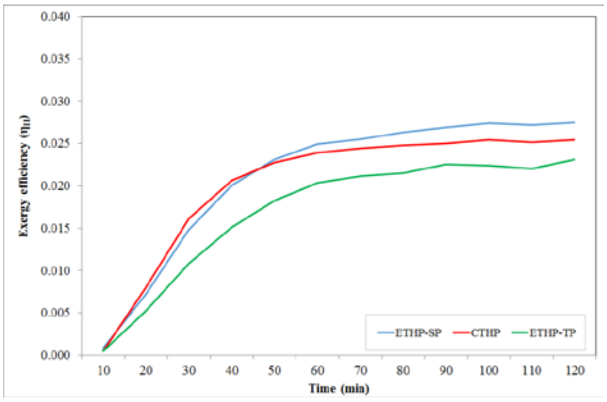

Fig. 9. Exergy efficiency values of the THPs for $V_{air} = 3.3\ ms^{-1}$

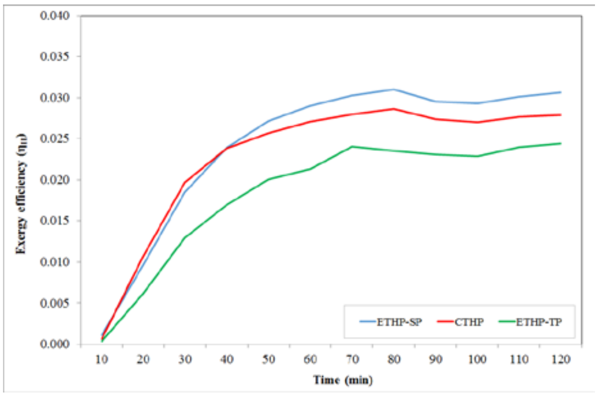

Fig. 10. Exergy efficiency values of the THPs for $V_{air} = 5.3\ ms^{-1}$

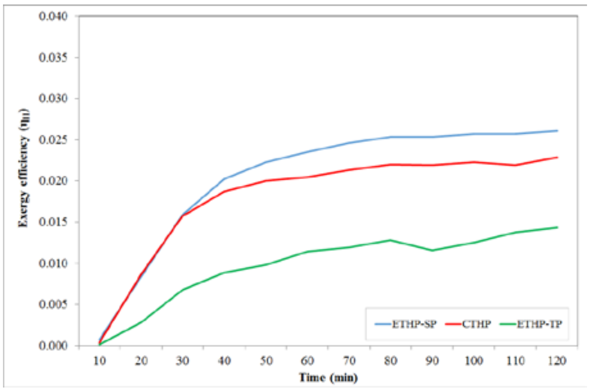

Fig. 11. Exergy efficiency values of the THPs for $V_{air} = 6.3\ ms^{-1}$

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	ETHP-SP			CTHP			ETHP-TP		
rate (ms^{-1})	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

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

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

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

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

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

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

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

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

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

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

- 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.

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