

AN EXPERIMENTAL INVESTIGATION OF THE THERMAL PERFORMANCE OF TWO-PHASE CLOSED THERMOSYPHON USING ZIRCONIA NANOFUID

by

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Original scientific paper
DOI:10.2298/TSCI140403116T

In this work an avant-garde study is made to find out the thermal performance of two-phase closed thermosyphon (TPCT) using ZrO_2 (5-25 nm) nanoparticle with base fluid water. The TPCT is fabricated from the copper tube with the outer diameters of 6 mm, 8 mm and length of 800 mm. The concentration of nanofluid is varied from 3% and 4% by weight. The experimental work is carried out under different operating conditions. The input parameters such as effects of input power, fill ratio and nanoparticle concentration on the thermal efficiency and thermal resistance of the TPCT were investigated, compared and discussed. From the results, it is observed that the efficiency of the TPCT increases when ZrO_2 /water nanofluid was used instead of water.

Key words: *two-phase closed thermosyphon, ZrO_2 nanofluid, thermal efficiency, thermal resistance*

Introduction

Heat pipe is a heat transport device functioning by the use of charged working fluid under vacuum condition. The heat pipe consists of three sections; evaporator, adiabatic and condenser section, fig.1(a). In the evaporator, heat is absorbed by the working fluid and it evaporates as it absorbs an amount of heat equivalent to the latent heat of vaporization. In the condenser, the vapor form of working fluid loses its heat and condenses into liquid and then, returns to the evaporator by means of capillary action in the wick. Normally, conventional fluids like water, ethanol and acetone are used to transport heat from one place to other. In addition to the conventional fluids, nanofluids are used to increase the speed of heat transfer. A very small amount of guest nanoparticles, when suspended uniformly and stably in base fluids, can provide dramatic improvement in working fluid thermal properties when compared with conventional working fluids without nanoparticles. The presence of nanoparticles is used to increase the thermal conductivity and convective heat transfer performance of the working fluid. A thermosyphon is similar to a heat pipe but without a wick. In the condenser, working fluid loses heat and returns to the evaporator by means of gravity, because the evaporator must be located vertically below the condenser, fig.1(b). In the present scenario, increasing the heat transfer rate

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using electronic equipment becomes a major problem. Heat pipes are used to increase the heat transfer in electronic products. The thermosyphon is used in various applications such as solar collector, heat exchangers, electronic devices cooling, and in the field of chemical engineering, waste heat recovery and so on. Numerous investigations have been conducted to study the thermal performance of heat pipes using nanofluids.

The limiting factor against increasing the heat transfer performance of heat pipe depends on the properties of the working fluid. The enhancement of liquid thermal conductivity is achieved by adding highly conductive solid nanoparticles within the base fluid. The special characteristics of the nanofluid substantially increase the heat transfer coefficient, thermal conductivity and liquid viscosity. Choi [1] proposed the concept of nanofluids first. Shafahi *et al.* [2] investigated the thermal performance of cylindrical heat pipes using Al_2O_3 , CuO and TiO_2 nanofluids. The two dimensional analysis was used to study the thermal performance of cylindrical heat pipe utilizing nanofluids, reported by Zhu and Vafai [3]. The velocity, pressure, temperature and the maximum heat transfer limit for the heat pipe for various operational conditions using nanofluid was carried out.

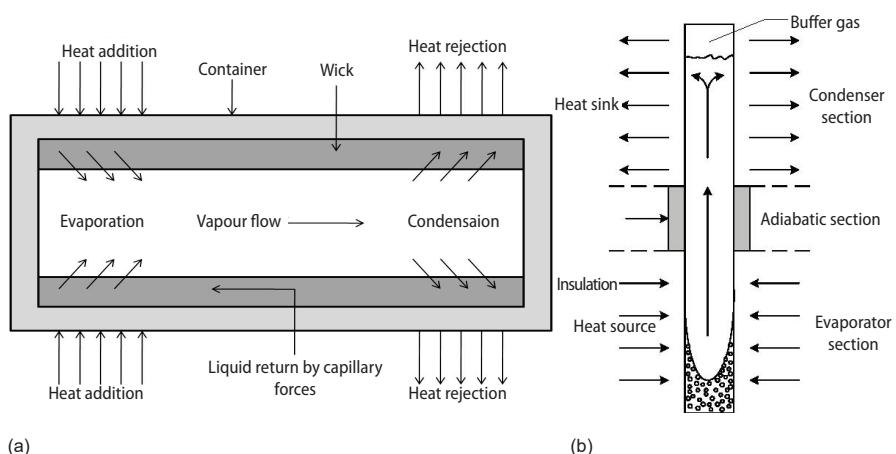


Figure 1. The construction and the principle of operation for (a) heat pipe, and (b) thermosyphon

Their finding shows that the thermal resistance decreases with an increase in concentration and a decrease in particle diameter. The existence of an optimum mass concentration and particle smaller in size providing the highest thermal performance has been recognized.

A comparison of the effects of measured and computed thermo physical properties of nanofluids on heat transfer performance were stated by Duangthongsuk and Wongwises [4]. The TiO_2 nanoparticles with average diameters of 21 nm and a volume fraction of 0.2-1 vol.% were used. Transient hot wire apparatus and Bohlin rotational rheometer were used to measure the thermal conductivity and viscosity of nanofluids, respectively. The results revealed that all the well-known correlations under estimate the thermal conductivity and viscosity of nanofluids compared with the measured data. Their finding shows that the transport behaviors of nanofluids were mainly dependent on the thermophysical properties of nanofluids.

Duangthongsuk and Wongwises [5] calculated the thermal and physical properties such as specific heat, viscosity and thermal conductivity of the nanofluids. The various models for predicting the thermo physical properties of nanofluids were summarized and used to calculate the experimental convective heat transfer coefficient of the nanofluid flowing in a

double-tube counter flow heat exchanger. TiO_2 nanoparticles with 0.2 vol.% dispersed in water were used. The result showed that there is no significant effect on the predicted values of the heat transfer coefficient of the nanofluid in case of very low level of concentration.

Naphon *et al.* [6] examined the thermal performance of heat pipe using titanium nanofluids. The fluid transport properties and flow features of the working fluid determine the heat transfer enhancement in heat pipes. The working fluids used de-ionic water, alcohol and nanofluids (alcohol and nanoparticles). The experimental set-up consists of a test section, refrigerant loop, cold-water loop and data acquisition system. The closed loop consists of a 0.3 m³ storage tank, an electric heater controlled by adjusting the voltage, and a cooling coil immersed in a storage tank. The results showed that the heat pipe thermal efficiency increases with an increase in heat flux and charge amount of working fluid.

The convective heat transfer and pressure drop of TiO_2 + water nanofluid through a horizontal circular tube premeditated by Kayhani *et al.* [7]. The diameter and volume concentrations of TiO_2 nanoparticles used in this study were 15 nm and 0.1, 0.5, 1.0, 1.5, and 2.0%, respectively. The results showed that the heat transfer coefficient and Nusselt number of the system in turbulent flow increases with increasing the nanofluid volume fraction. For TiO_2 -water nanofluid with 2% volume fraction, the Nusselt number increased by 8% at $\text{Re} = 11,780$ and there was no significant increase in pressure drop for the nanofluid.

The surface temperature and vapor temperature of an air-cooled condenser heat pipe at steady and transient condition have been determined by Annamalai and Ramalingam [8]. The results bared the truth that the performance of condensing process is affected by the low surface convective heat transfer coefficient in the condenser. Their finding shows that the water-cooled condenser can be used to enhance the performance of the heat pipe. The comparison of temperature distribution and the heat transfer rate of the thermosyphon heat pipe with iron oxide nanofluid and DI water, were considered by Huminic *et al.* [9].

Huminic and Huminic [10] offered the heat transfer characteristics of two-phase closed thermosyphon [TPCT] with iron oxide nanofluid. It is attributed to the fact that the heat transfer rate increases with an increase in inclination angle caused by using nanofluid and also by dispersing higher concentration of iron oxide nanoparticles in pure water. The thermal resistance decreases with the increases in inclination angle and it also increases in volume concentration. The heat pipes filled with self-rewetting fluids have more stability, higher thermal efficiency and lower thermal resistance than heat pipe filled with water suggested by Senthilkumar *et al.* [11]. The results displayed those aqueous solutions of n-pentanol give better results than the aqueous solution of n-butanol.

Williams *et al.* [12] examined the turbulent convective heat transfer behaviour of alumina (Al_2O_3) and zirconia (ZrO_2) nanoparticle dispersions in water, in a flow loop with a horizontal tube test section at various flow rates. The result showed that there was no abnormal heat transfer enhancement. Kim *et al.* [13] deliberated the pool boiling characteristics of dilute dispersions of alumina, zirconia and silica nanoparticles in water. Their finding shows that a significant enhancement in critical heat flux (CHF) can be achieved at modest nanoparticle concentrations (<0.1% by volume). Rea *et al.* [14] probed the laminar convective heat transfer and viscous pressure loss for alumina-water and zirconia-water nanofluids in a flow loop with a vertical heated tube. The outcomes showed that the zirconia-water nanofluid heat transfer coefficient increases by approximately 2% in the entrance region and 3% in the fully developed region at 1.32 vol.%.

Only very few authors attempted to study the thermal behavior of ZrO_2 nanofluids in heat pipes and TPCT. In this work, it is introduced to study the effect of ZrO_2 nanofluid as an

absorbing medium on the thermal performance of TPCT. This study is different from the previous investigators, using TPCT with two different diameters (6 mm and 8 mm).

Description of experiment

Nanofluid preparation and evaluation of properties

The ZrO₂ nanoparticles were purchased from Reinst Nano ventures, Pune, India. To prepare nanofluid, the nanoparticles with required mass concentration of 3% and 4% by weight were dispersed in water followed by a stirring action done by ultrasonic homogenizer for 90 minutes to break any possible aggregations of nanoparticles and to keep the nanofluids uniformly dispersed. The reason for choosing 3% and 4% concentrations may be that the thermal performance of TPCT increases with an increase in concentration of nanoparticles in base fluid. The thermo physical properties of the nanoparticle are shown in tab.1. No surfactant was used.

Table 1. Thermophysical properties of ZrO₂ nanoparticles

Particle	Average diameter [nm]	Purity	Specific surface area [m ² g ⁻¹]	Thermal conductivity [Wm ⁻¹ K ⁻¹] of nanofluid at 30 °C		
				Concentration	Measured value	Predicted value
ZrO ₂	5-25	>97.2%	130 ± 20	0%	0.6	0.6
				3%	0.631	0.6336
				4%	0.687	0.6425

Thermophysical properties including thermal conductivity, the density, the specific heat, and the viscosity of nanofluid have been introduced by Yang and Liu [15]:

- density of ZrO₂ nanofluid (ρ_{nf})

$$\rho_{nf} = \phi\rho_s + (1-\phi)\rho \quad (1)$$

- effective thermal conductivity of dilute nanofluid (k_{nf})

$$\frac{k_{nf}}{k} = \frac{k_s + 2k + 2\phi(k_s - k)}{k_s + 2k - \phi(k_s - k)} \quad (2)$$

- specific heat of the nanofluid ($c_{p,nf}$)

$$(\rho c_p)_{nf} = (1-\phi)(\rho c_p)_s + \phi(\rho c_p)_s \quad (3)$$

- viscosity of the dilute nanofluid (μ_{nf})

$$\mu_{nf} = \mu(1 + 2.5\phi) \quad (4)$$

The thermal conductivities of nanofluids were measured by using a KD2 Pro thermal property analyser by Suresh *et al.* [16].

Heat pipe construction

Heat pipe is fabricated from straight copper tubes with outer diameters of 8 mm for the condenser section, and 6 mm for the evaporator section. The purpose of 8 mm diameter is to increase the surface area to enhance the heat transfer in the condenser section. The thickness and length of the TPCT used in these experiments are 1 mm and 800 mm, respectively. The distribution of the thermocouples on the heat pipe tested is indicated in fig. 2.

Experimental set-up

The schematic diagram of the experimental set-up is shown in fig.3. The set-up consists of a test section, digital temperature indicator, ammeter, voltmeter, heater coil, autotransformer, thermocouples, etc. A straight copper tube 800 mm long, 250 mm, 220 mm and 330 mm long for evaporator, adiabatic and condenser section respectively, are used as a test section. Calibrated K-type thermocouples of ± 0.5 °C are placed in the surface of the test section to measure the outside wall temperature. Two thermocouples are attached to the evaporator; the other thermocouples are attached to the adiabatic and condenser section, two thermocouples are used to measure the inlet and outlet temperature of the water cooled condenser. Coil type heater (maximum 500 W) is used as a heat source in the heating section. The terminals of the heater coil are attached to an auto-transformer using which heat flux can be varied by varying the voltage. The thick insulation is provided over the heater coil and adiabatic section uses glass wool to minimize heat loss.

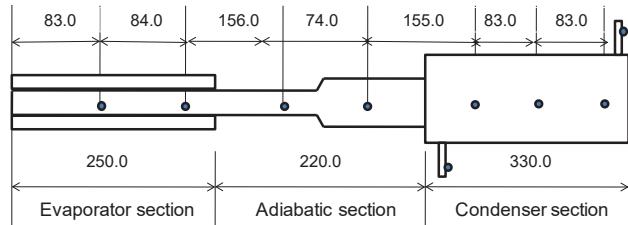


Figure 2. Thermocouples distributions on the tested TPCT

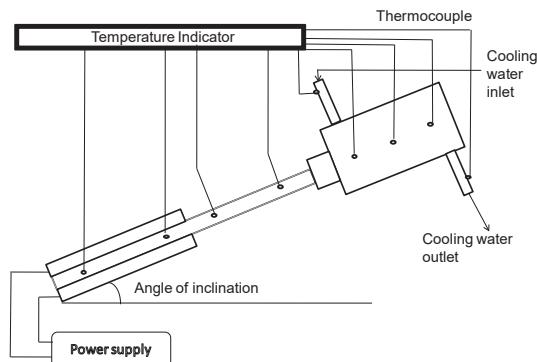


Figure 3. Experimental set-up and thermocouples distributions on the tested TPCT

Experimental procedure

The procedure is initiated when the vacuum requirements are achieved using vacuum pump. The working fluid is charged into the evacuated pipe. The supplied electric power is adjusted manually at the desired rate using the auto transformer. At this point in the tests, it took approximately 35 to 40 minutes to reach steady-state. TPCT is insulated with glass wool of 20 mm thickness. The surface temperatures of the evaporator and condenser section, the mass flow rate of the cooling water are also measured. It is assumed that under steady state condition, the wall temperature is approximately equal to the working fluid flowing inside the heat pipe. Once the steady-state condition has been reached, the temperature distribution along the TPCT was measured and recorded, sequentially using the selector switch and along with the other experimental parameters. The local temperature on the TPCT is measured by seven isolated K-type thermocouples. Thus, the heating load (Q) and temperature difference were measured, and the thermal resistance (R) of TPCT is calculated by eq. (14). In the present study, the TPCT was fixed at 90° during measurement. The input power is increased incrementally, and the process is repeated for various parameters.

Data reduction and uncertainty analysis

From the measured data, the following quantities are calculated [17].

Inlet heat by evaporation of a TPCT is defined by:

$$Q_{in} = VI - Q_{loss} \quad (5)$$

where Q_{loss} is the total heat loss from the evaporator section by radiation and free convection as follows:

$$Q_{\text{loss}} = Q_{\text{rad}} + Q_{\text{conv}} \quad (6)$$

$$Q_{\text{rad}} = \varepsilon \delta A (T_{\text{ins}}^4 - T_{\text{surr}}^4) \quad (7)$$

$$Q_{\text{conv}} = h_{\text{conv}} A (T_{\text{ins}} - T_{\text{surr}}) \quad (8)$$

Free convection heat transfer coefficients can be determined using the equation [18]:

$$\text{Nu} = \frac{h_{\text{conv}} L_t}{k_{\text{surr}}} = \left\{ 0.825 + \frac{0.387 (\text{Ra})^{1/6}}{\left[1 + \left(\frac{0.492}{\text{Pr}} \right)^{9/16} \right]^{8/27}} \right\}^2 \quad (9)$$

$$\text{Ra} = \frac{g \beta (T_{\text{ins}} - T_{\text{surr}}) L_t^3}{\alpha g} \quad (10)$$

$$\text{Pr} = \frac{\vartheta}{\alpha} \quad (11)$$

Outlet heat by condensation of a TPCT is defined by:

$$Q_{\text{out}} = m c_p (T_{\text{out}} - T_{\text{in}}) \quad (12)$$

The efficiency of TPCT is defined by:

$$\eta = \frac{Q_{\text{out}}}{Q_{\text{in}}} \quad (13)$$

Thermal resistance of a TPCT is defined by:

$$R = \frac{\Delta T}{\dot{Q}_{\text{max}}} \quad (14)$$

where the temperature difference between the mean temperature of condenser and evaporator is:

$$\Delta T = \frac{T_{e1} + T_{e2}}{2} - \frac{T_{c1} + T_{c2} + T_{c3}}{3} \quad (15)$$

Uncertainty of the experimental data resulted from measuring errors of parameters such as current, voltage, mass flow rate and inlet and outlet temperature of cooling water can be calculated using the relations for efficiency [19]:

$$\max E_{\eta} = \pm \sqrt{(E_{Q_{\text{out}}})^2 + (-E_{Q_{\text{in}}})^2} \quad (16)$$

$$\max E_{Q_{\text{out}}} = \pm \sqrt{(E_m)^2 + (E_{c_p})^2 + (E_{(T_{\text{out}} - T_{\text{in}})})^2} \quad (17)$$

$$\max E_{Q_{in}} = \pm \sqrt{(E_V)^2 + (E_I)^2} \quad (18)$$

Because of small order of magnitude, the effect of Q_{loss} on uncertainty can be neglected. The maximum precision of ammeter and voltmeter was 0.1 A and 0.5 V, respectively. An uncertainty analysis of measured and calculated quantities has been done and the uncertainty values are presented in tab. 2.

Results and discussions

To evaluate the thermal performance of the system, the following ranges of operating conditions were fixed for testing the experiment:

- the heat pipe tilt angle was 90°,
- the input power to the evaporator was varied from 10 W to 100 W,
- the fill ratio, which is the ratio of the volume of charged fluid to the total evaporator volume was fixed at 0.5, and
- the concentration of ZrO₂ nanoparticle were kept at 3% and 4% by weight.

Figures 4(a)-(d) presents the temperature distribution of the wall of the TPCT for the various input power at an inclination of 90°. For higher input powers, the average temperatures

Table 2. Uncertainties in the measured and calculated quantities

Measured/calculated quantity	Uncertainty
Temperature	±0.5 °C
Mass flow rate	±5.0 %
Heat input	±5.1 %
Heat output	±5.23 %
Efficiency	±5.38 %
Thermal resistance	±2.87 %

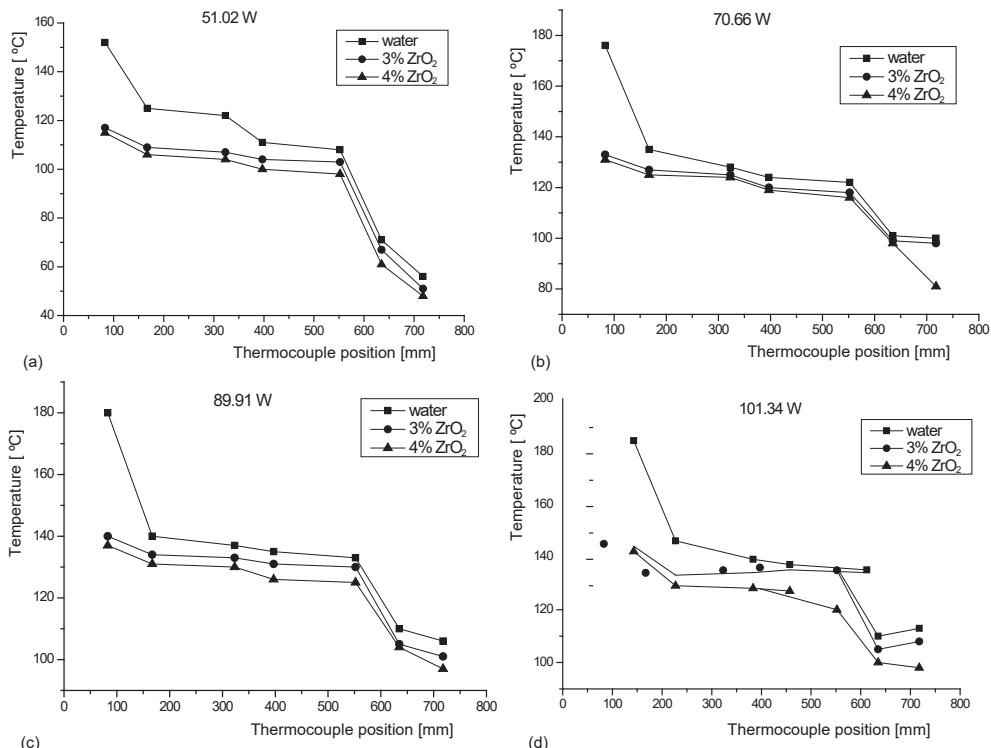


Figure 4. Average surface temperature of TPCT vs. various input power and concentration of nanofluid

within sections of TPCT increase. The thermal performance of TPCT is better when ZrO_2 nanofluid is used instead of pure water. This may be due to the temperature difference between evaporator and condenser section which is lower than pure water. The range of surface temperature seems to be lower with 4 wt.% ZrO_2 nanofluid compared to 3 wt.% ZrO_2 nanofluid and pure water. This is attributed to the fact that increasing the concentration of nanoparticle in the fluid increases the conductive heat transfer along the tube and decreases the convective heat transfer across the wall.

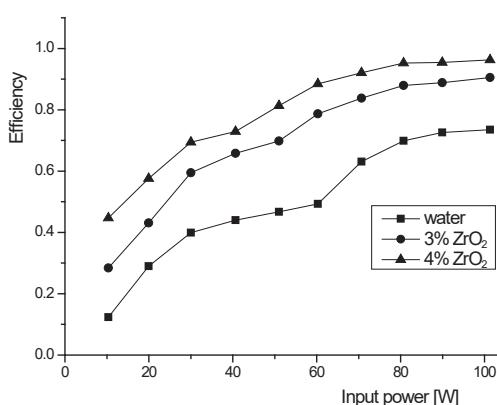


Figure 5. Efficiency of TPCT vs. input power and concentration of nanofluid

observed to be high in case of 4 wt.% zirconia nanofluid compared to that of other fluids irrespective of the input power. This correlates with the observations on temperature distribution at the wall surface.

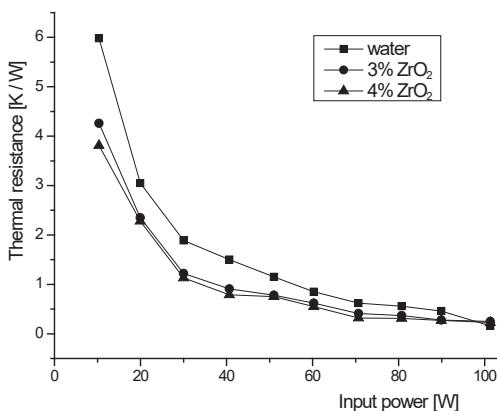


Figure 6. Thermal resistance of thermosyphon vs. input power and concentration of nanofluid

to the evaporator section. On the other hand, these thermal resistances reduce suddenly to the minimum value when the heat load is increased.

Figure 7 shows the efficiency of TPCT vs. average surface temperature of evaporator section for different input powers. The efficiency of TPCT enhances with increasing input power and

The heat pipe thermal efficiency can be evaluated by finding the ratio between the heat removed at the condenser section and the heat supplied at the evaporator section. For different input powers (51.02-101.34 W), the efficiency was calculated and is presented in fig. 5. When the TPCT is charged with nanofluids, the efficiency is significantly enhanced. For example, at the input power of 89.91 W, 3% nanofluid can improve the efficiency of the TPCT from 72.59% to 88.82%. This improvement increases with increase in mass concentration of nanoparticle and input power. The thermal efficiency of the heat pipe is less at lower input power, due to the poor evaporation rate of working fluid in the evaporator. The thermal efficiency was

The thermal resistance of the heat pipe (R_{th}) is defined as the ratio between the change in temperature between the evaporator section and the condenser section and the amount of heat supplied to the evaporator section, eq. (4). Figure 6 shows the variation of thermal resistance of the TPCT related to various input powers for zirconia nanofluid and pure water. It was observed that the thermal resistance of heat pipe decreases for all combinations of working fluid with increasing value of the input power. It is due to the fact that the surface temperature of the zirconia nanofluids is less than that of the water *i.e.* more amount of heat is carried away by the zirconia nanofluids in the evaporator section. However, the thermal resistance of heat pipes using both the base fluid and nanofluid are comparatively high, due to more liquid film resides in the evaporator section.

decreasing evaporator average surface temperature. Increasing the concentration of nanoparticles to water led to improvement in the efficiency of TPCT and decreasing the evaporator average temperature is shown in the figs. 4(a)-(d) and 5. The thermal performance enhancement in TPCT by nanofluid mainly depends on particle size, particle type, particle shape and base fluid.

Conclusions

An experimental investigation on TPCT by using zirconia nanofluid was carried out. Different mass concentrations of nanoparticles (3-4%) in suspension within the TPCT were experimentally examined and results were compared with pure water. The following conclusions can be drawn based on the obtained results.

- The thermal performance of TPCT is better with nanofluids in all concentrations than water. The efficiency of TPCT enhanced upto 22.8%.
 - The thermal resistance of the TPCT decreases with an increase in nanoparticle concentration.
- This finding shows nanofluid appealing as working medium in TPCT. Observations of further investigations are needed.

Nomenclature

A	- external surface of insulate tube, [m^2]	Ra	- Rayleigh number, [$= \beta(T_{\text{ins}} - T_{\text{surr}})L_t^3/\alpha\vartheta$], [10], [-]
c_p	- specific heat of water, [$\text{Jkg}^{-1}\text{K}^{-1}$]	T_c	- condenser temperature, [$^\circ\text{C}$]
g	- acceleration due to gravity, [ms^{-2}]	T_e	- evaporator temperature, [$^\circ\text{C}$]
h_{conv}	- convective heat transfer coefficient, [$\text{Wm}^{-2}\text{C}^{-1}$]	T_{in}	- inlet temperature of cooling water, [K]
I	- current, [A]	T_{ins}	- temperature on external surface of insulation, [K]
k	- thermal conductivity of base fluid, [$\text{Wm}^{-1}\text{K}^{-1}$]	T_{out}	- outlet temperature of cooling water, [K]
k_{nf}	- thermal conductivity of nanofluid, [$\text{Wm}^{-1}\text{K}^{-1}$]	T_{surr}	- surrounding temperature ,[K]
k_s	- thermal conductivity of nanoparticle, [$\text{Wm}^{-1}\text{K}^{-1}$]	ΔT	- temperature difference, [K]
k_{surr}	- thermal conductivity of surrounding air, [$\text{Wm}^{-1}\text{C}^{-1}$]	V	- voltage, [V]
L_t	- total length of tube, [m]		
m	- coolant water mass rate, [kgs^{-1}]		
Nu	- Nusselt number ($= h_{\text{conv}}L_t/k_{\text{surr}}$), [9], [-]		
Pr	- Prandtl number ($= \vartheta/\alpha$), [11], [-]		
Q_{conv}	- convection heat transfer rate, [W]		
Q_{in}	- inlet heat by evaporation, [W]		
Q_{loss}	- heat loss by radiation and convection, [W]		
Q_{max}	- maximum heat transfer rate, [W]		
Q_{out}	- outlet heat by condensation, [W]		
Q_{rad}	- radiation heat transfer rate, [W]		
R	- thermal resistance, [KW^{-1}]		

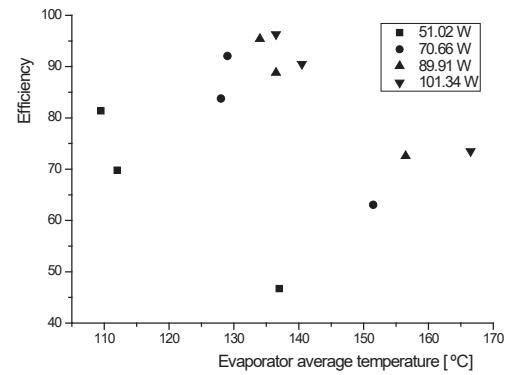


Figure 7. Efficiency of TPCT vs. average temperature of evaporator for different input powers

Greek symbols

α	- thermal diffusivity [m^2s^{-1}]
β	- coefficient of thermal expansion [K^{-1}]
δ	- Boltzmann constant in eq. (7), [$\text{Wm}^{-2}\text{K}^{-4}$]
ε	- emissivity factor of insulator, [-]
ϑ	- thermal diffusivity, [m^2s^{-1}]
η	- efficiency of TPCT, [-]
μ	- viscosity of base fluid, [Nsm^{-2}]
μ_{nf}	- viscosity of nanofluid, [Nsm^{-2}]
ρ	- density of base fluid, [kgm^{-3}]
ρ_{nf}	- density of nanofluid, [kgm^{-3}]
ρ_s	- density of nanoparticle, [kgm^{-3}]
ϕ	- nanoparticle mass fraction, [-]

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