# EXPERIMENTAL STUDY OF A PELTIER COOLER COUPLED TO A THERMOSIPHON HEAT SINK

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*This paper describes an experiment investigation of Peltier water cooler that modified by adding a thermosiphon to enhance heat dissipation from hot junction. The thermosiphon is worked by natural circulation and working three different are tested to explore its effectiveness, which water, kerosene, and ethylene glycol solution (50/50 mixture of distilled water). Experimental work is conducted using domestic Peltier water cooler with a 760 g water tank capacity. The thermosiphon unit is added as a modification to the traditional finned heat sink connected to fan. Two different arrangements of thermosiphon are investigated, one with single riser tube, while the other has two riser tubes. The results indicates that the design with single riser tube show stable operation and gives efficient heat dissipation rather that the dual riser tube. The findings also revealed that kerosene is the best fluid to use in increasing the coefficient of performance in the system.*

Keywords: *Thermosiphon, heat sink, Peltier cooler, natural circulation, COP*

## 1. Introduction

One of the modern types of cooling equipment is the thermoelectric cooler (TEC) or Peltier cooler, which is fastest developing. These systems work on the principle of Peltier effect, which takes place when an electric current is made to flow between two semiconductor junctions resulting in the direct movement of heat on the cold surface to the hot one. It is not a secret that the lower the temperature of the heat sink, the higher the coefficient of performance (COP) [1,2]. In clean energy as well as heat dissipation applications where components require low operating temperatures, the buoyancy-driven natural circulation systems play a very essential role in various thermal systems [3]. The fact that a thermosiphon can be used to extract waste heat without the use of mechanical pumping and since it has a low thermal resistance because of its effective convective heat transfer is two valuable benefits. The applications of thermosiphons are very numerous, and they are commonly used within the thermal management of nuclear reactors, geothermal energy recovery, cooling of electronic devices, and solar power networks [4,5]. Some of the benefits of thermoelectric coolers over traditional compressor-based systems are that they are smaller, generate less noise, and that they do not use refrigerants. The TEC that is chosen in this study is the popular semiconductor-based thermoelectric modules of cooling electronic devices [6]. When an electric current is passed through two sets of semiconductors, the heat

is conducted off the cold side to the hot side producing the effect of cooling in Peltier conductors. It is a well-known fact that the decreasing temperature of the hot side increases the COP of the system. The fact that this type of cooling technology operates quietly and is independent of orientation is one of its most significant advantages. The primary limitation of TEC, however, remains the relatively low COP (smaller than one) that reduces its overall energy efficiency [7]. One of the greatest challenges in development of TEC is efficient heat removal. The thermal design and heat transfer characteristics enhance the cooling capability and the general performance of TEC systems significantly. This improvement is often achieved by incorporating heat dissipation structures to optimize thermal management, or incorporating auxiliary materials, including phase change materials [8]. Heavy heat is being extracted out of miniature electronic components more and more by thermosiphons. Thermosiphons have become more of an important factor in the preservation of efficient heat dissipation as electronic devices have continued to demand increased power densities and smaller sizes. A typical thermosiphon consists of three key parts: a condenser, adiabatic part and evaporator. A working fluid is introduced in it to fill it to some extent through buoyancy forces. Once the heat is absorbed and released the fluid in the evaporator then ascends to the condenser and condenses into fluid state. This liquid is then recirculated to the evaporator becoming a natural circulation cycle [9,10]. Such circulation is caused by the forces of buoyancy due to the difference in density between the heated and cooled sections. The low thermal resistance of thermosiphons and the fact that they do not require the use of mechanical pumps makes them easy and efficient in heat transfer. Thermosiphons are thus common in industrial applications in the geothermal energy systems, solar collectors, and electronic cooling. The thermo-physical properties of the working fluid, especially its density, viscosity and specific heat capacity, have a significant effect on the performance of a thermosiphon [11]. The high power consumption and the use of refrigerants make the traditional vapor-compression refrigeration systems popular in cooling water have a major adverse environmental effect. Alternatively, TEC is a feasible alternative, as it proposes an environmentally-friendly approach towards thermal control. In the section that follows, a synopsis of important research that has been conducted to examine the effectiveness of Peltier coolers with modified heat sinks is given.

Vián *et al.* [12] explored a home TEC together with a thermosiphon. Two samples were experimented where one was equipped with a traditional finned heat dissipater sample and the other equipped with a thermosiphon to reject heat. These findings indicated that the addition of the thermosiphon enhanced the COP by as much as 6.42% compared to the regular design. In order to improve the work of the TEC systems more, Riffat and Ma [13] considered the optimization of module design, employment of state-of-the-art thermoelectric materials, and the high heat exchange performance on the hot and cold surface. Their findings indicate that the maximum COP is the highest when water cooling on the hot side is used as compared to the rest of the configurations that have been tested. To store foods at 5 ℃, Vián and Astrain [14] designed a TEC with one compartment, which had 0.225 m3 in size. They showed that adding two-phase thermosiphons and capillary lift mechanism to the system increased the COP by 66 percent compared to the conventional finned heat sinks. On the same note, Sun *et al.* [15] tested in their experimental a TEC that uses gravity-assisted heat pipe as a heat sink on the hot side. Their findings revealed that such an alteration minimized the electricity usage by 42.20% and enhanced the cooling capacity by approximately 73.54. The effect of placement of thermoelectric modules on the cooling performance was an experimental study carried out by Mirmanto *et al.* [16]. The three possible module positions on the side wall, under the box and top side were tested.

The study findings indicate that the thermoelectric module mounted on the side wall has the most desirable cooling effects. Winarta *et al.* [17] examined the efficiency of U-shaped heat pipe filled with a working fluid that is methanol at varying filling ratios. Their findings demonstrated that filling ratios of 45 and 55 percent, respectively, gave the maximum values of COP of 0.03881 and 0.03885. Abderezzak *et al.* [18] suggested new thermoelectric cooling system which involves finned and flat heat pipes. They obtained the following results: The new arrangement produced a COP of 0.74 and a temperature difference of 32.7 ℃. The article of Aqeel *et al.* [19] introduced a theoretical study of the impact of heat sink geometry on the thermoelectric performance. They tested squares, plate and circular fin designs and discovered that longer fin design boosts the COP. The best of all designs that were considered was the rectangular fins. Lertsatitthanakorn *et al.* [20] experimentally studied a thermoelectric dehumidification system integrated with a heat pipe as the heat sink. Their findings revealed that the use of heat pipes enhanced both the COP and cooling capacity. Similarly, Jamradloedluk *et al.* [21] developed a thermoelectric water cooler integrated with a heat pipe heat sink. When compared with a conventional water cooler, the proposed system achieved a higher COP, despite a slightly higher initial capital cost. More recently, Kang et al [22] introduced a mini TEC that incorporating a radial heatsink designed especially for LED systems. The new configuration enhanced both cooling effect and system efficiency. The obtained experimental results showed that this modification achieved 1.81 times higher in COP in active mode by lowering and over 50 % lower thermal resistance by 50% in thermal resistance compared to original conventional designs. The modifications of hot side in thermoelectric applications were not limited solely to the use for cooling applications, but similar developments were also made in the case of using the thermoelectric principle as power generators (TEG). For example, Gu *et al.* [23] explored the enhancement of heat collection for TEG through the use of micro heat pipe arrays attached at the hot end. They quantified the joint actions of the bending angles, length and inclination among other structural parameters. Comparative analysis reveals that proposed configuration could achieve up to a 47.27% increase in power output compared to conventional arrangement. Majed and Moghimi [24] presented a numerical study of a TEG coupled to thermosiphon system that filled with nanofluids. The theoretical study investigated how fin geometry and nanofluid type affect the performance. Results showed that optimizing fin height and spacing significantly enhances heat dissipation.

Previous studies on TEC systems have primarily focused on improving the performance of the Peltier module or enhancing heat dissipation through conventional finned heat sinks. However, limited attention has been given to integrating passive cooling methods such as thermosiphons with TEC to improve heat rejection efficiency together with conventional active methods (finned heat sink and fan). Moreover, the influence of thermosiphon configuration and working fluid type on the overall cooling duty remains insufficiently studied. Therefore, the main aim of this paper is to explore the thermal performance of TEC in which the hot junction is cooled by a combined finned heat sink with fan and thermosiphon. Three different working fluids are tested: water, kerosene, and a 50/50 ethylene glycol–water solution, and two thermosiphon configurations (single and double riser tubes) are investigated.

## 2. Methodology

Fig. 1 illustrates a sketch of the experimental device used in the present study. The test rig is done in the Basra Engineering Technical College at the thermal mechanical department lab. The specifications of the test rig items were shown in Tab. 1.

### *2.1. General Description*

The experimental work was carried out based on a small-sized Peltier water dispenser that contains a water tank of 760 g capacity. The basic design of the cooler consists of the water tank, a solid aluminum block, a thermoelectric module, an aluminum finned heat sink, and a dissipated fan. The first proposed modification made to the cooler is to add a thermosiphon on the hot side of the thermoelectric module, which is backed by the same original arrangement, i.e., finned heat sink and the fan. The first proposed design of the thermosiphon consists of a fine polished aluminum water box, a single riser tube, a finned condenser unit, and a downcomer tube. The second proposed design of the thermosiphon consists of the same thermosiphon components but with dual riser tubes.

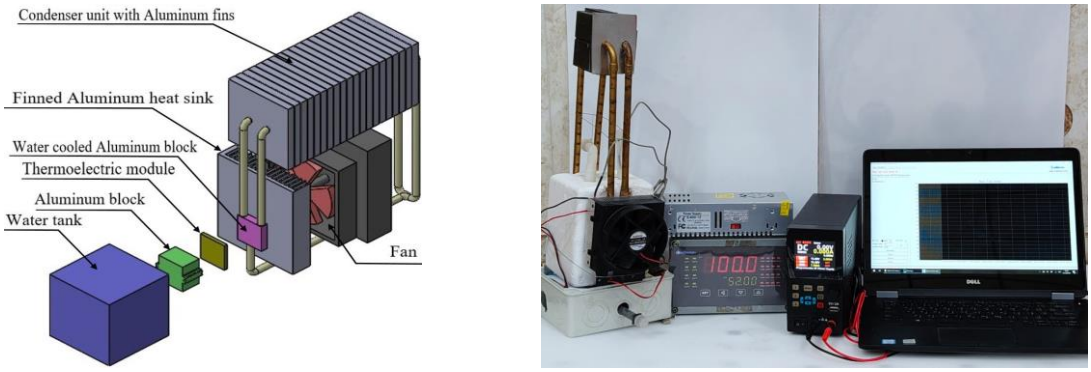


**Fig. 1 Schematic and image of the experimental test rig**

**Table 1. Specifications of the test rig items**

| Item | Specification |
|---|---|
| Water storage tank | -Plastic material of capacity 760g; -Insulated by 2-cm of foam. |
| Solid Aluminum block | Dimension $(4 \times 4 \times 2) cm$ |
| Thermoelectric module | -Type: TEC1-12706 of rated voltage: 12 V, (40 × 40 × 4 mm) |
| Water-cooled aluminum block | -Dimensions $(4 \times 4 \times 2) cm$<br>-Single input port and single or double output ports. |
| Riser tube | -Copper material. (D=1cm, L=28cm) |
| Condenser unit | -Single or dual copper pipe D=1cm<br>-Aluminum fins (20fins, (8×8×1 cm) |
| Downcomer | Single or dual copper pipe (D=1cm, L=28cm) |
| Finned heat sink | -Aluminum material; -Base: 100× 100 mm.<br>-28-Fins: 100×20 mm; -Gap between fins: 2mm |
| External fan | Axial type of 2 W power |

### *2.2. Measuring instruments and controllers*

To judge the performance enhancement that could be attained from the modification to the TEC, several measuring instruments and controls are used in this work, including:

1- K-type thermocouple wires to measure the temperatures at designated locations in the test rig.

2- Digital temperature data logger to measure the output signal from thermocouple wires. To guarantee accuracy, calibration is done against a mercury thermometer for the selected channels.
3-DC digital ammeter and voltmeter to measure the electrical power consumed by the TEC module.
4- Power supply of 12 V DC to power the thermoelectric module.

The specifications of the selected measuring instruments and controllers are given in Tab. 2.

**Table 2.** Specifications of the measuring instruments

| Item | Type/Model | Resolution | Accuracy | Range |
|---|---|---|---|---|
| Thermocouple wires | K-type | - | - | -200-1260℃ |
| Temperature data logger. | Model: MPD-580 | 0.1°C | ±0.2% | -50 to +1200°C |
| Digital Ammeter/voltmeter | Brand: WOWOONE | 0.01 A<br>0.1 V | ±1% | 0-10 A<br>0-100V |
| D.C. power supply | Model: ATO-S-300-12<br>Voltage:12V; Current: 25A | - | - | - |

### *2.3. Experimental Procedure*

Initially, the test rig was operated without any modifications. Electrical DC power is supplied to the thermoelectric module, and heat begins to transfer from the water contained in the storage tank towards the finned heat sink. The forced convection produced by the fan will dissipate the heat energy to the surroundings. After completing this procedure, the hot side of the thermoelectric module is modified first by adding a single riser tube. The final procedure includes modifying the thermosiphon by adding double riser tubes instead of a single one to assess its effects on performance. A high-quality thermal adhesive was applied to minimize contact resistance to the lowest possible levels between the various components of the test rig. Three fluids are used in the thermosiphon, which are water, kerosene, and ethylene glycol (EG). Various properties of the selected fluids are given in Tab. 3. The performance of the TEC was judged before the modification through the measured temperatures and DC electric power consumption. The readings were done until the temperature approached 15 ℃, as this value is the set point of a thermostat unit. In order to extent the data acquired from the rig, the stored water is preheated to 40 ℃ before starting the test. Sampling rate is set to 2 min.

**Table 3. Properties of Thermosiphon working fluids at 25°C.**

| Property | Water | Kerosene | Ethylene glycol (EG) |
|---|---|---|---|
| Density(g/cm³) | 0.997 | 0.79 | 1.081 |
| Dynamic Viscosity[mPa·s] | 0.89 | 1.7 | 3.5 |
| Thermal Conductivity,[$Wm^{-1}$ $K^{-1}$] | 0.606 | 0.16 | 0.4 |
| Specific Heat Capacity, (kJ $kg^{-1}$ $K^{-1}$) | 4.18 | 2.1 | 3.4 |
| Boiling Point@1 atm [℃] | 100 | 230 | 107 |

### *2.4. Data Calculations*

The power consumed by the thermoelectric module $Power_{in}$, is calculated by:

$$Power_{in} = V * I \tag{1}$$

Where $V$ is the applied voltage (about 12 V), and $I$ is the consumed current.

The input heat energy to the module from the storage water tank is calculated using:

$$Q_{in} = m_w \, Cp_w \, \frac{dT}{dt} \tag{2}$$

Where: $m_w$ is the mass of water contained in the tank (=0.760 kg). ; $Cp_w$: is the specific heat of water =4.18 kJ $kg^{-1}K^{-1}$. $dT/dt$: is the temperature gradient with time. The value of the temperature gradient is calculated using the measured cold-water temperature. So, by using the least squares method, the variation of cold-water temperature is represented by [25]:

$$T = a + bt \tag{3}$$

Where $a, b$ are constants. Applied conservation of energy (first law of thermodynamics) on the thermoelectric module allows for computing the output of the thermal energy that must be dissipated to the air as [26]:

$$Q_{out} = Power_{in} + Q_{in} \tag{4}$$

The coefficient of performance (COP) of the system is given by [26]:

$$COP = \frac{Q_{in}}{Power_{in}} \tag{5}$$

### *2.5. Uncertainty Analysis*

The value of the COP of the TEC is found with respect to measurement of cooling effect and power consumption. Evaluation of the uncertainty associated with this measurement requires the uncertainty associated with temperature measurement, voltmeter, ammeter, and balance (to measure mass in the cold tank). The error associated with COP is calculated by the root sum square method as:

$$e_{COP} = \sqrt{\left(\frac{\partial(COP)}{\partial m} e_m\right)^2 + \left(\frac{\partial(COP)}{\partial b} e_b\right)^2 + \left(\frac{\partial(COP)}{\partial A} e_A\right)^2 + \left(\frac{\partial(COP)}{\partial V} e_V\right)^2} \tag{6}$$

The error associated with the temperature gradient $b$ depends on the error associated with the reading of each temperature sensor as well as the error associated with the measurement of the time. Generally, it is given by:

$$e_b = \sqrt{\left(\frac{\partial b}{\partial T_1} e_{T_1}\right)^2 + \cdots + \left(\frac{\partial b}{\partial T_n} e_{T_n}\right)^2 + \left(\frac{\partial b}{\partial t_1} e_{t_1}\right)^2 + \cdots + \left(\frac{\partial b}{\partial t_n} e_{t_n}\right)^2} \tag{7}$$

## 3. Results and Discussion

A TEC system based on the Peltier effect requires efficient heat dissipation to achieve best cooling performance. Enhancing the heat removal process is essential for improving the overall COP of the system. To address this, we adopted a thermosiphon that works through natural circulation, utilizing three different working fluids. This section presents and discusses the experimental results observing the influence of a closed-loop thermosiphon on the performance of the TEC. Results are reported for

experiments using water, kerosene, and an ethylene glycol solution as recommended working fluid to fill the thermosiphon.

### *3.1 Temperature distribution without thermosiphon*

In this section, the performance of the TEC without the thermosiphon was evaluated. The hot side of the TEC unit consisted of an aluminum heat sink equipped with an air fan to dissipate heat. Initially, the water in the storage tank was preheated to approximately 40 °C, as previously described. Fig. 2 illustrates the variation of the cold- and hot-side temperatures over time in the absence of the thermosiphon. The recorded data show that the cold-side temperature decreased linearly over a period of about 148 minutes, while the hot-side temperature gradually increased to around 56 °C. During this period, the system consumed approximately 41.06 W of electrical power, as shown in Fig. 3. A slight fluctuation was observed during the first 10 minutes, attributed to the initial preheating of the water, which influenced the main thermoelectric module's stability. Under these conditions, the COP of the TEC was determined to be 0.212±3.02%.

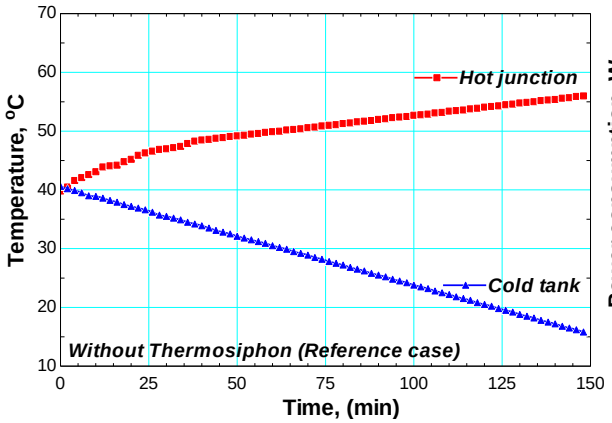


**Fig. 2 Temperatures variation with time for the reference case**

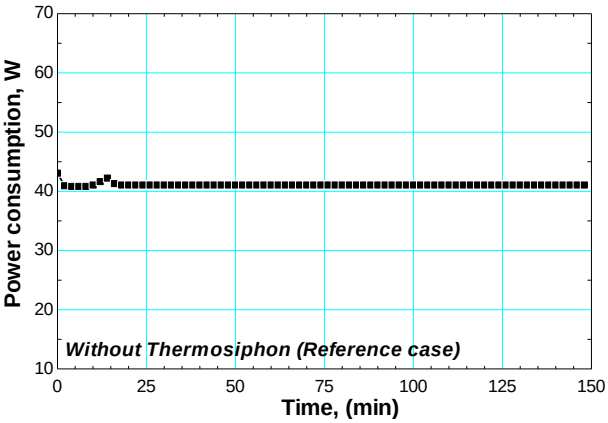


**Fig. 3 Variation of power consumption with time for the reference case**

### *3.2 THC with a thermosiphon and water as working fluid*

Two new types of heat sinks were designed to replace the traditional finned heat sink: a single-tube thermosiphon and a two-tube thermosiphon. These modifications were investigated to assess their potential for enhancing the cooling capacity of TEC systems. Fig. 4(A) presents the variation of water temperature in the tank, the temperature of the hot junction, and the temperature of the finned heat sink over time after incorporating the single-tube thermosiphon. As shown, the temperature of the stored water decreases smoothly and nearly linearly, similar to the trend observed before the modification. At the beginning of operation, the hot junction temperature rises sharply during the first 20 minutes, reaching about 55 °C, before gradually decreasing and stabilizing at approximately 45.5 °C. This initial increase is primarily due to the lack of natural circulation within the thermosiphon at low temperature differences, when the density gradient between the liquid and vapor phases is small. Since buoyancy-driven flow in a thermosiphon relies on density variations, insufficient stratification at the start delays circulation. As heating continues, the water in the evaporator region becomes less dense, while the cooler liquid in the condenser section remains relatively denser. This increasing density difference may cause

the working fluid to circulate naturally due to the stable convective loop that is initiated by it. When circulation is developed the temperature at the hot junction is reduced since the well-known regime of convective heat transfer replaces the earlier regime dominated by conduction. The viscosity decreases with temperature, besides helping to ensure a smoother flow via the loop, flow resistance decreases, and circulation stabilizes. Moreover, the water thermal conductivity is rather high (0.6 W/m.K at room temperature) that enhances the rate of heat transfer in the condenser and evaporator regions, making the process of the efficient exchange of energy and stabilization of temperature easier. Better natural convection and effective thermal conduction gives a more homogenous distribution of temperature along the thermosiphon. More so, the temperature of the hot junction is reached sooner since the already heated water accelerates the boiling and vapor generation process of the evaporator part. When a steady circulation has been established, the fan-assisted finned heat sink is useful to hold the condenser temperature at a steady value of about 34 ℃, which means that the change creates an almost constant heat removal process. The use of the single-tube thermosiphon provided a 82 minute thermal response time and an overall COP of 0.388+ 2.12 %. This is better than the reference configuration and proves the useful nature of the good thermos-physical properties of water, namely, high thermal conductivity, moderate viscosity, and high density-led buoyancy. Combined, these characteristics enhance constant circulation and enhanced heat dissipation.

Fig. 4(B) shows the observed changes in temperature on changing the system to run on two pipes as compared to when it operated on a single pipe. As it is evident, the cooling time of the stored water also doubled and was approximated to 164 minutes, compared to a response time of 82 minutes in a single tube setup. Though the modification was made, the temperature of the finned heat sink was not significantly different in comparison with the single-tube case. This means that the heat dissipation rate on this side was not very sensitive. The heat sink temperature increased to approximately 35 ℃, but no longer, indicating that the design change did not make a significant difference to the external convective and fan-aided cooling conditions. However, the behavior of the hot junction exhibited a distinct difference. The hot junction temperature showed a more pronounced rise, peaking at approximately 60 °C after 30 minutes of operation. This value is significantly higher than the steady-state value observed in the case of single-tube system. This behavior can be attributed to the delayed of natural circulation that causing partial stagnation of the working fluid within the dual thermosiphon risers. In the two-tube arrangement, the flow distribution between the parallel tubes depends on temperature gradient and thereby local density difference. Because water's viscosity and density both vary strongly with temperature, even a small asymmetry between the two tubes can cause unbalanced circulation. Consequently, a single tube can regulate the flow leaving the other tube stationary. The onset of circulation is further inhibited by the viscosity at the onset of heating (approximately 40 ℃). The low-density difference between the hot bottom layer and the cold top layer slows down the motion caused by buoyancy. This unstable circulation also leads to intermittent oscillations in flow, which increases the thermal resistance of the loop. As a result, the hot junction temperature remains at high temperatures longer before stabilizing and the thermosiphon works more inefficiently. Once circulation finally advances, the system reaches a quasi-steady state, but with lower effectiveness compared to the single-tube case. The temperature difference between the hot and cold sections indicates that heat removal was hindered by partial fluid stagnation and reduced convective mixing. Consequently, the COP decreased markedly to 0.168±3.19%. In summary, while the two-tube thermosiphon design was intended to increase heat transport capacity, the interaction between water's thermophysical properties,

specifically its temperature-dependent viscosity, density, and thermal conductivity beside the flow distribution dynamics led to unstable and inefficient circulation.

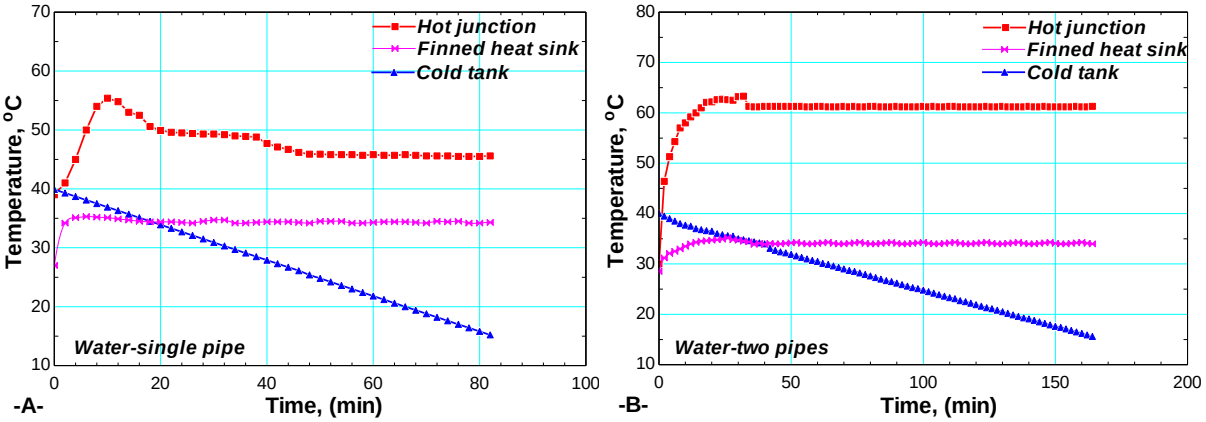


**Fig. 4 Variation of temperatures with time for the modified TEC, -A- single riser tube thermosiphon, -B- two riser tubes thermosiphon, both using water as a working fluid.**

A comparison of the electrical energy consumed by the TEC with thermosiphon system using water as the working fluid is shown in Fig. 5. As illustrated, the response time is considerably faster in the single-tube design than in the double-tube one. The figure also shows that the single-tube system operates under more favorable conditions in terms of thermal stability and energy consumption. In the single-tube case, the average electrical energy consumption was approximately 41.1 W, with only minor fluctuations during the initial startup phase. These initial oscillations represent the period prior to the steady circulation of the fluids being fully developed.

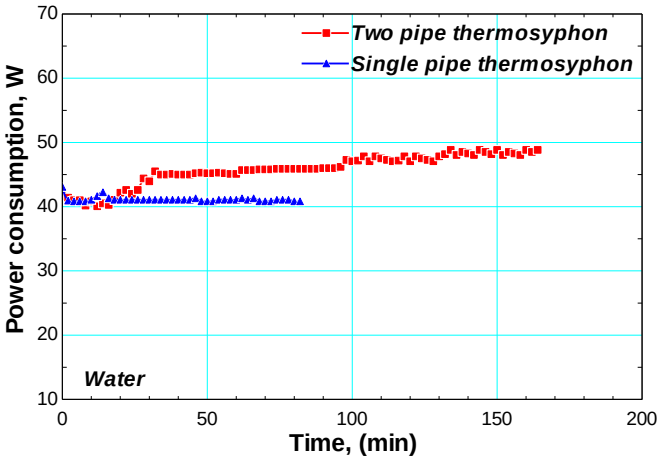


**Fig. 5 Variation of power consumption for the single-tube and double-tubes riser thermosiphon using water as a working fluid**

Once the stabilization of the power demand was established, the power demand was also stable, thus suggesting that the TEC was functioning effectively. Conversely, the double-tube thermosiphon model revealed significant deviation of power consumptions which continued almost 30 minutes after which power averaged around 45.7 W. This unbalanced fluid circulation of the two parallel risers increases the higher energy draw and delayed stabilization. The TEC has to work at different loads of thermal loads due to the intermittent stagnation and uneven distribution of flow in one of the tubes and makes the heat transfer at the hot junction intermittent. Such oscillations raise the mean electric energy

use and extend the cooling reaction time as the module settles to non-resonant heat elimination. This also leads to the fact that the double-tube setup is less energy-efficient and less thermal duty than the single-tube system. The positive thermophysical characteristics of water make it possible to circulate continuously, promoting more similar heat dissipation and reduce the need to use electricity.

### *3.3 TEC with a thermosiphon and kerosene as working fluid*

Fig. 6(A) presents the temperature variations of the thermosiphon unified-tube working fluid with kerosene as a working fluid. The findings indicate that the TEC responds well and similarly to thermal conditions, and it takes about 56 minutes to stabilize to steady-state operation. This is an outstanding enhancement of the reference case where there is no thermosiphon. The cold-water temperature shows a slow and almost linear decrease but the hot junction temperature remains constant and close to 44 ℃. The key factor that allows one to observe no noticeable changes in the temperature of the hot junction is the fact that kerosene is characterized by moderate viscosity and relatively low density that make it possible to initiate the natural circulation rather quickly. The difference in density between the hot and cold legs of the thermosiphon is smaller and thus reduces the inertia of the fluid and accelerates the formation of a convective loop that is in a stable state. Also, since kerosene has a relatively low specific heat capacity, it has a higher ability to heat faster as it promotes the commencement of circulation. Although the thermal conductivity is lower than that of water, the continuous and stable flow compensates for this limitation by maintaining consistent convective heat transfer. The finned heat sink temperature remained close to previous case, confirming that the external heat dissipation rate did not vary significantly with the change in working fluid. Overall, this arrangement achieved a COP of 0.5460.168±1.29%.

The performance of the two-tube thermosiphon using kerosene as the working fluid is shown in Fig. 6(B). Similar to the behavior observed in case of water, this configuration produced less satisfactory results. Firstly, the response time increased substantially to 102 minutes, almost twice that of the single-tube case. Secondly, the hot junction temperature displayed a more unstable trend, with obvious fluctuations before stabilizing after about 52 minutes at approximately 50.5 °C. This instability can be explained by the uneven distribution of flow between the two parallel riser tubes. Because kerosene has higher viscosity and a lower density than water, the buoyancy forces causing natural circulation are weaker, while viscous resistance is more pronounced. When the flow is divided between two risers, small temperature or pressure imbalances cause one tube to dominate the circulation while the other becomes partially stagnant. This oscillatory flow behavior increases the thermal resistance of the assembly, increasing the hot junction temperatures and slowing the cooling response. It has been shown that the fan-assisted finned heat sink was able to maintain constant heat rejection over a temperature range of just 28 to 35℃ on the surface. As a result, the calculated COP reduced to 0.246±2.89%.

Fig. 7 shows the variation in electrical power consumption of the two setups used with kerosene as the working fluid. The findings explain perfectly the extent of performance improvement of the single-tube thermosiphon. It had a constant power consumption as long as it was operational. Conversely, there were significant changes in the power consumption in the first 30-40 min with the double-tube design because circulation and redistribution of flows between the two risers were irregular. The time in which the cooling process occurred was almost twice in the case of the two tubes, which demonstrates how unstable natural convection lowers thermal and electrical efficiency. The stable flow regime of the single-tube configuration ensured reduced power usage, more uniform heat transfer and

enhanced cooling performance. In general, the thermosiphon with a single tube showed a balanced circulation and low power consumption when the working fluid was kerosene. Conversely, the double-tube system minimized thermal duty because it amplified hydraulic resistance and produced an asymmetrical flow.

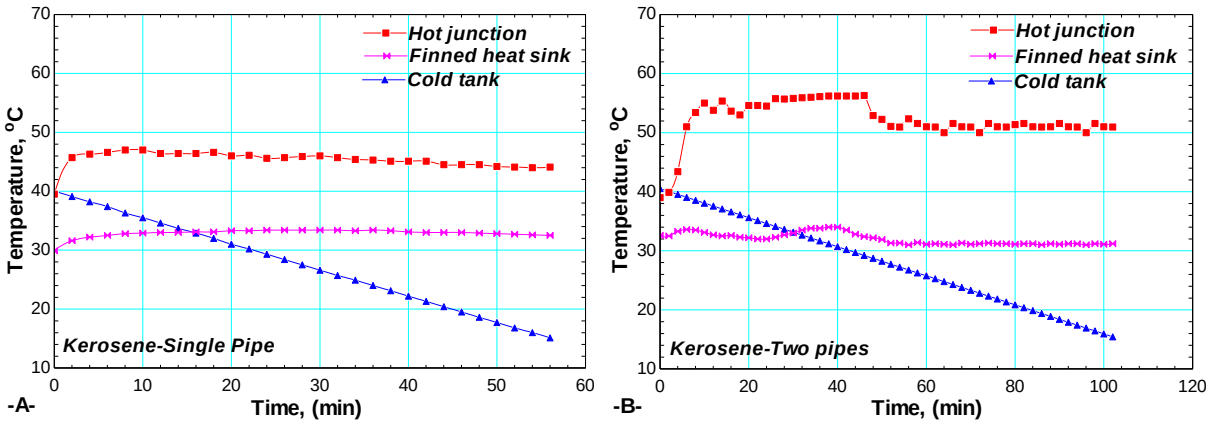


**Fig. 6 Variation of temperatures with time for the modified TEC, -A- single riser tube thermosiphon, -B- two riser tubes thermosiphon, both using kerosene as a working fluid.**

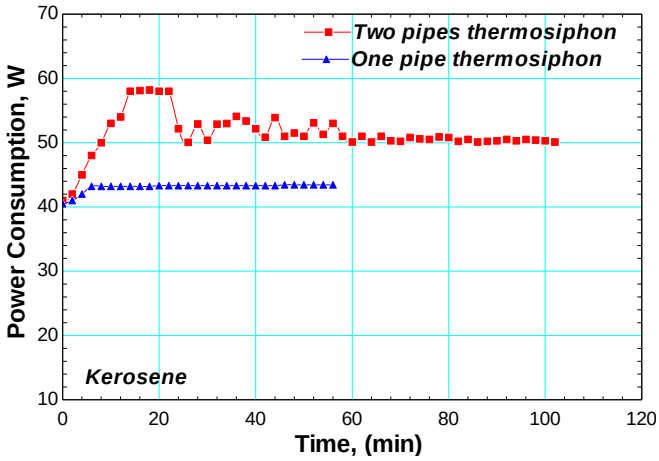


**Fig. 7 Variation of power consumption for the single and double riser tubes thermosiphon using kerosene as a working fluid**

### *3.4 TEC with a thermosiphon and ethylene glycol solution as working fluid*

Fig. 8(A) shows the temperature variation in the system after solution of ethylene glycol was placed as the working fluid with a thermosiphon with one riser tube. It is also apparent that the system has a long response time- of about two hours. The linear temperature distribution in the stored water will remain constant and gradually decrease until the desired temperature is achieved. The perceived challenge of the circulation of fluids in the initial stage of operation is underscored by the figure of the hot junction temperature that reaches a maximum of 57.5 ℃ in the minute 48. This is due to the fact that the working fluid is highly dense, necessitating the existence of a high temperature gradient to generate effective circulation. As can be seen by the above illustrations, the finned heat sink temperature does not vary appreciably, but there is a small increase at startup due to the elevated temperature of the hot junction. The COP of this case was determined to be 0.255±2.99%.

Fig. 8(B) represents the temperature changes of the thermosiphon in the presence of two riser tubes and ethylene glycol being the working fluid. Remarkably, the response time of this design is quite similar to that of the single-tube one; it is approximately 126 minutes to achieve the desired temperature; this is only six minutes longer than the former configuration. The temperature of the hot junction is also varied during the startup as there is no circulation and the temperature reaches its highest point at 60 °C at the sixteenth minute. This elevated temperature is attributed to the low thermal conductivity of the working fluid, which increases conduction resistance. As circulation stabilizes, the temperature gradually decreases and settles around 53 °C. Based on the corresponding calculations, the COP for this case is determined to be 0.225±3.09%.

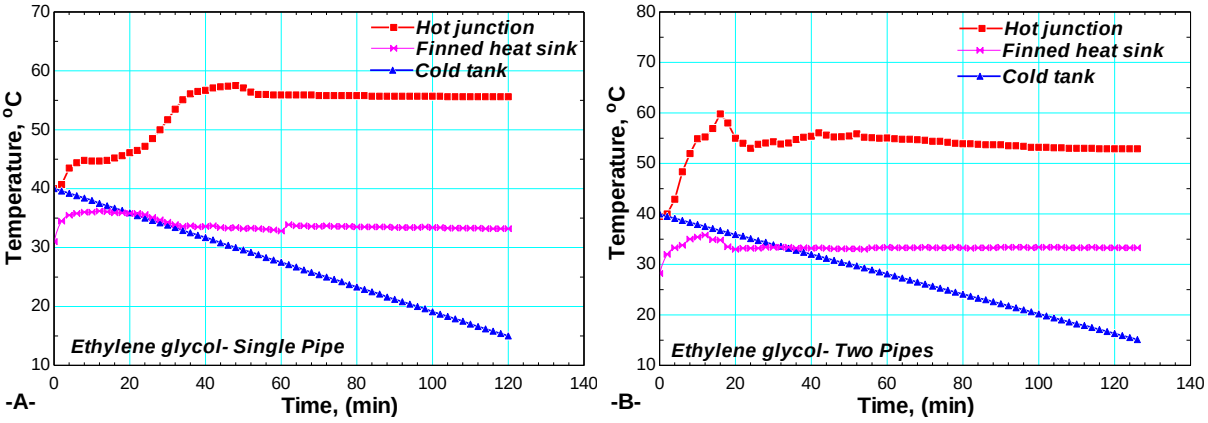


**Fig. 8 Variation of temperatures with time for the modified TEC, -A- single riser tube thermosiphon, -B- two riser tubes thermosiphon, both using ethylene glycol as a working fluid.**

Fig. 9 illustrates the variation in energy consumption of the TEC system when ethylene glycol is used as the working fluid for both proposed thermosiphon designs. The results show a strong similarity between the two cases, with a slight increase observed in the double-tube configuration, particularly during the start-up phase around the 30th minute.

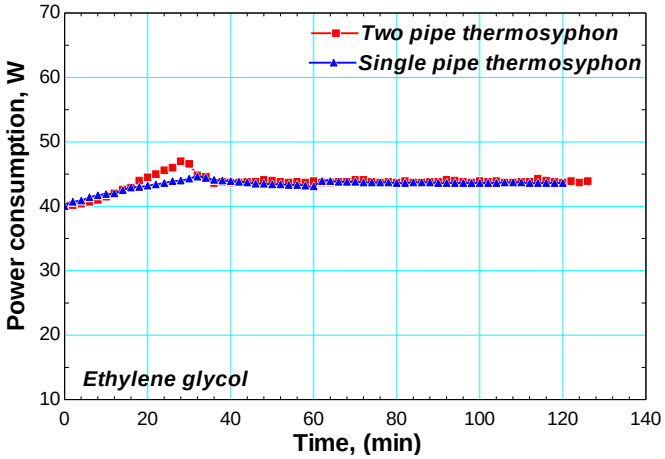


**Fig. 9 Variation of power consumption for the single and double riser tubes thermosiphon using ethylene glycol solution as a working fluid**

### *3.5 Performance comparison of TEC with and without a thermosiphon*

Fig. 10 compares the COP of TEC systems employing different working fluids: water, ethylene glycol solution, and kerosene, in single-pipe, two-pipe, beside siphon-free configurations. The results indicate that the single-tube configuration filled with kerosene exhibits the highest thermal efficiency among all tested cases, achieving a COP of 0.546. Conversely, the kerosene double-tube design exhibits a lower COP, suggesting that the addition of a second riser tube may result in increased flow resistance and thermal inefficiency rather than increased heat transfer efficiency. The COP for the single-tube design using water as the working fluid was 0.388, which is less than that of kerosene but better than the performance of the ethylene glycol solution. The conclusion that a dual-tube design might lower system efficiency was supported by the double-pipe design filled with water, which displayed a notable drop in COP. In both single-tube and double-tube designs, the ethylene glycol solution showed the lowest COP values of any tested fluid, indicating that it is not very suitable as a working fluid in the current operating environment. No matter what working fluid is used, the results generally demonstrate that the system has the lowest COP when a thermosiphon is absent, underscoring the critical role that the thermosiphon plays in enhancing overall cooling efficiency. A comparison of the current results with those reported in previous studies shows that Vián and Astrain [14] achieved a 66% improvement in COP by incorporating two-phase thermosiphons and capillary lift into a TEC. Recently, Kang et al [22] reported a more significant enhancement of 81% through design modifications on the hot side. In the present study, the performance improvement reached 158%, which substantially exceeds the gains reported in earlier works.

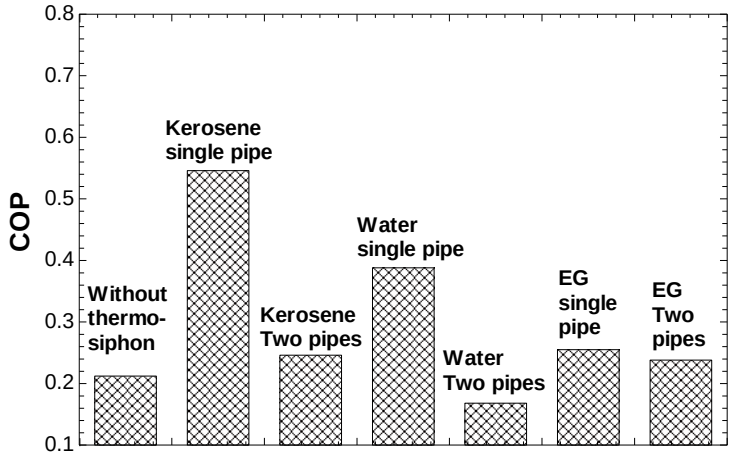


**Fig. 10 COP values for the Peltier cooler**

### *3.6 Limitations and challenges of TEC with a thermosiphon*

While the incorporation of a thermosiphon device with the TEC improves performance by enhancing heat dissipation, it unfortunately introduces certain limitations. The thermosiphon device is a gravity-assisted, passive heat transfer device, and its performance is strongly dependent on orientation. This requirement reduces the portability and orientation independence that are often regarded as advantages of stand-alone TEC. Consequently, the combined system may not be suitable for applications where compactness, portability, or arbitrary mounting positions are critical. In addition, the added size and complexity of the thermosiphon may limit the feasibility of integration into small-scale or mobile cooling systems.

## 4. Conclusions

A Peltier cooler coupled to a modified heat sink was experimentally tested for this study. The modification involved adding a thermosiphon to a traditional finned heat sink. The study examined the effects of geometric variations in the thermosiphon when using different working fluids, including water, kerosene, and ethylene glycol (EG) mixed in a 50/50 ratio with distilled water. The study's findings included the following conclusions:

1. By reducing the hot-side temperature, the thermal gradient across the module is increased, improving the overall performance and cooling efficiency of a Peltier cooler.
2. TEC performance can be improved by adding a thermosiphon to a conventional finned heat sink, depending on the geometry design and working fluid.
3. Because two riser tubes tend to show flow instability, a thermosiphon with a single riser tube performs better than one with two.
4. Kerosene was determined to be the optimal working fluid based on the thermosiphon's current dimensions, which led to a 158% increase in the refrigerator's COP.

It is advised that the riser tube length be changed for subsequent research because it could affect the thermosiphon's observed performance for each working fluid.

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**Nomenclature**

| | | | |
|---|---|---|---|
| $Cp_w$ | Specific heat of water, [kJ kg$^{-1}$ K$^{-1}$] | $Q_{out}$ | Dissipated heat, [W] |
| COP | Coefficient of performance[-] | $Q_{in}$ | Cooling effect, [W] |
| $I$ | Electric current, [A] | $t$ | Time, [sec] |
| $m_w$ | Mass of water, [kg] | $T$ | Temperature, [℃] |
| $Power_i$ | Input electrical power [W] | $V$ | Voltage, [V] |

RECEIVED DATE: 7.6.2025.
DATE OF CORRECTED PAPER: 22.9.25.
DATE OF ACCEPTED PAPER: 27.10.25.