Experimental verification for the phases separation technique to improve the thermal performance of vertical and inclined wickless heat pipe

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Abstract. A flow separation technique is tested experimentally for the thermal performance enhancement of the Thermosyphon Heat Pipe (THP). Flow separation for the vapor rising upward and the condensate flowing downward is used to eliminate the complex interactions between the two phases. Such interactions are found and discussed by previous numerical work published by the authors. Tubes Packing (TP) made of two parallel copper tubes welded contrary to each other with a neighboring openings at central disk are used to grantee the smooth flow through the THP. The THP is made with a suitable dimension for the evaporator, adiabatic and condensation sections. Different operating conditions of filling ratios, supplied heat and inclination angles are tested to monitor the THP thermal performance during transient and steady state operation. The experimental tests results showed that the insertion of TP into the THP enhanced the thermal performance of THP by reducing the transient operation time about 37% and the maximum averaged temperature in the evaporator section about 50%. Also, reduces the variation effects of the filling ratio and the inclination angle on the thermal performance. In general, the results showed that the insertion of the TP reduces the THP thermal resistance about 17% to 62% based on the operating conditions.

1.Introduction

Heat pipes are one of the modern heat transfer devices which is a full vacuumed closed pipe consist of three sections named evaporator, adiabatic, and condenser [1], [2]. Heat pipe is partially filled with a working fluid (de-ionized water in this study). Heat pipe is a passive heat transfer device due to cyclic evaporation and condensation of the working fluid that make it as a super conductor for heat absorbing from hot surfaces or spots [3], [4]. Heat pipes are widely used in different of applications such as: Cooling of electronic devices [5], enhancement of thermal dissipation from electronic components [6], enhancement of thermal control of thermal storage systems [7], enhancement of energy saving performance of chip cooling [8], absorb and transfer high rates of solar energy in solar panels [9] [10], air conditioning applications [11], heat ventilation and air conditioning systems HVAC [12], space and satellite missions [13], cooling of wind tower [14], heat recovery and renewable energy applications [15], thermal management of coal fields [16], thermal management of gas turbine blades disk [17]. Therefore, heat pipes are considered as a device that used to solve the high rates heat transfer problems or at least enhance the heat transfer rates from relatively high temperature zones.

The heat pipe is divided, based on working principles, into three sections: Evaporator section, Condenser section and adiabatic section [2]. In the evaporator section the heat is absorbed from the source. In the condenser section the heat is released to the cooling media. The adiabatic section connects evaporator and condenser sections and considered as a fully insulated section.
When the working fluid absorbs heat from the inner surface of the evaporator section and reaches its boiling temperature. A phase change from the liquid phase to the vapor phase will occur [18]. This evaporation process is accompanied with high rates of heat absorbing from the heat supply source. The amount of the absorbed heat depends on the amount of the generated vapor and latent heat of evaporation of the working fluid [19]. The vapor travel to the condenser section through adiabatic section. At inner wall of the condenser section, the vapor suffering another phase change process from vapor phase to liquid phase. This condensation process accompanied by high rates of heat releasing from the vapor to the cooling media. The condensate returns to the evaporator section and continues the evaporation / condensation processes cycle. For straight pipe, the condensate returns to the evaporator section with the aid of a capillary structure (wick) as in wick heat pipe, or by gravity forces for wickless heat pipe. Wickless heat pipes are normally known as Thermosyphone Heat Pipes (THP) and its working cycle is clarified in figure 1. [20].

![Figure 1. Classical schematic diagram of THP [20]](image)

The THP is the basic type of all other HP types that developed to ensure reliable returning of condensed liquid to the evaporator section. Therefore, the enhancement of THP thermal performance can enhance other types thermal performance. Researchers are doing their best to enhance THP thermal performance in many different techniques such as Binary Mixture Technique that depends on the wettability of binary mixture of fluids. These fluids have an enhanced adhesion and surface tension forces that reduces the entrainment effects during the travelling of the two phases streams inside the THP as studied by H. Ahmad , 2011 [21] , M. Karthikeyan et.al ,2011 [22] , Y. Naresh et.al. , 2018 [23] and Y. Hu et.al. , 2018 [24]...etc.

Different techniques are used to enhance the thermal performance of the THP such as the use of phase change materials as investigated by Behi et.al., 2017 [25]. The use of nanoparticles techniques as studied by Kim et.al. 2019 [26]. The use of binary mixtures techniques as investigated by Naresh et.al., 2018 [23]. The fourth category followed by the researchers to enhance the thermal performance of the THP is the Geometry Techniques that depends on the geometrical aspects of the THP. Some of
these researchers are as follows: Y. Hung et.al. 2011 [27], developed a 1D mathematical model to study the effect of star grooves, HP cross-sectional area, HP length, adiabatic section length and FR on heat transfer capacity of micro HP. The star grooves types are: triangle, square, hexagonal, and octagonal. They concluded that for thin grooves the heat transfer capacity was increased due to the increases of capillary forces. The heat transfer capacity was proportionality to the cross sectional area and inversely to the length of HP. The heat transfer was increased when adiabatic section length decreased at constant HP total length. R. Nair et.al. 2016 [28], studied numerically the effect of the insertion of a rectangular longitudinal fins (along condenser section) and the number of fins on enhancement of the thermal performance of THP. The tested variable is the condensate mass of THP at steady state conditions. The results showed that the condensate mass is increased about 22% for 8 fins, and about 32% for 12 fins. A. B. Solomon et.al. 2017 [29], studied THP thermal performance with/without a thin porous copper coat at the inner surface of a coper pipe. The thin layer of porous copper was established to enhance pool boiling process and was made using electrochemical deposition process. Also they tested the effect of coating with oxides on the thermal performance. They found that the copper coat shows better heat transfer capabilities better than oxide coat. The uncoated pipe $R_{th}$ was lowered about 21% at inclination angle of 45º and 10 kW/m² heat flux. Y. Naresh et.al. 2017 [30], tested experimentally the effect of the attaching longitudinal fins to the inner surface of the condenser on THP thermal performance. The results showed that $R_{th}$ was lowered by 32% at FR 50% and 50 W supplied power. S. Fertahi et.al. 2018 [31], set up a 2D CFD numerical simulation using ANSYS / Fluent software to simulate THP for domestic water heating system. They suggested that the insertion of tilted fins at condenser section will enhance the thermal performance of THP. A. Alammar et.al. 2018 [32], studied experimentally the effect of roughness of the internal surface of THP on the thermal performance. The results declared that the surface roughness was reduced the $R_{th}$ about 16%. Y. Naresh et.al. 2018 [33], studied numerically 1D fluid flow and experimentally the effect of attaching longitudinal fins to the inner surface of the condenser on THP thermal performance. The results showed that $R_{th}$ was lowered by 35% at FR 50% and 50 W. Y. Kim et.al. 2019 [26], tested experimentally the effect of sintered microporous coating of evaporator section on THP thermal performance. The results showed that the reduction in $R_{th}$ was about 51% at FR 35% and about 30% at FR 70%. The best performance was recorded at inclination angles between 15º and 30º from horizontal. A. Temimy et.al. 2019,2020 [20], [34], [35], publish a series of articles about the flow patterns of the two-phase flow inside the THP numerically and experimentally. The simulation show that the flow patterns are not classical as demonstrated in figure 1 but it’s a non-steady spatial flow pattern. Also they propose and simulate numerically a Tubes Packing (TP) to be inserted inside the THP to control the flow streams of both steam and condensate stream. The TP is made from two parallel copper tubes welded contrary to each other with a neighboring openings at central disk. The upper tube called Riser tube have 9.52mm OD and 300mm long control the flow of steam streams. The lower tube called Downcomer tube have 5mm ID and 225mm long control the flow of condensate streams. The 3DCFD results show that the insertion of the TP produce a uniform circulation of the two-phases. Also it can enhance/reduce the $R_{th}$ of the THP up to 55%.

The goal of this work is to verify experimentally the effect of the flow separation technique on the thermal performance of the THP. The effects of filling ratio, supplied heat and inclination angle with/without the TP on the thermal performance is also verified.

2. Experimental Work
In this study, experimental work represents the main goal for validation of 3DCFD model simulation results and checking the effect of TP on thermal performance of THP. The test rig is designed and constructed to run with the same testing variables described in 3DCFD simulation model that given by refs. [20], [34], [35]. The THP and TP model geometry and dimensions are shown in figures 2 and 3 respectively.
A 600mm long copper tube has 19.05mm/17.4mm OD/ID was selected as THP. At the lower opening, a tab with a threaded lock was welded. This tab is used for insertion of TP inside the THP. At the upper THP opening, three 5 mm ID copper tubes were welded. The first tube was for vacuum gauge connection through a threaded tab. Second tube with a one-way valve (inlet to THP) for injection of the working fluid. Third tube with a one-way valve (outlet from THP) for vacuuming. A 200mm long copper tube with 53mm/50mm OD/ID mounted as jacketed cooling heat exchanger around the condenser section with inlet (lower) and outlet (upper) nozzles of 5mm ID tubes.

Eleven thermocouples type K were positioned at the surface of the THP and cooling water inlet and outlet nozzles to get the temperature distribution along THP. The thermocouples were calibrated with standard calibration procedure using pre calibrated water thermostatic bath type GD 120-S26 (operating range of −15 °C to 120 °C and ±0.1 °C accuracy). The thermocouples distributed along THP to trace temperature variations in each section especially in evaporation section \[36\]–\[38\]. Five thermocouples are set to the evaporator section numbered (1,2,3,4,5). Three thermocouples are set to the adiabatic section numbered (5,6,7). Three thermocouples are set to the condenser section numbered (7,8,9).

Figure 2. The THP model geometry and dimensions \([20],[34],[35]\)

Figure 3. The Tubes Packing (TP) model geometry and dimensions \([20],[34],[35]\)
Twelve channels temperature recorder type BTM-4208SD (uncertainty ±0.5 °C) with 16G SD card were used to record the temperatures during the tests for transient and steady state operations. A transient recording step was set to 1 Sec. A schematic diagram for the test rig is shown in figure 4.

![Figure 4. Test Rig Schematic Diagram](image)

The heat is supplied by six cartridge cylindrical heaters of 50 Watt each were arranged with equal spacing and tightened with two screw clamps around the evaporator section. Also, to ensure that heat generated by the heaters is distributed uniformly to the evaporator surface. Special thermal clay was applied to fill the spaces between heaters and distribute the heating power uniformly. Finally, the THP body and thermocouples were covered with a one-inch thickness single layer of thermal insulation glass wool. Then wrapped with a thermal insulation tape.

Cooling water was arranged as one-way flow by gravity from an over-head 120 lit. tank. The cooling water tank was mounted with 2*1,500 W heaters with automatic temperature control system adjusted at 25 °C (the experimental tests are done in winter). The final test rig arrangement and components are shown in figure 5.
Figure 5. Test rig arrangement and Components

Two copper tubes were arranged and welded together parallel to each other with an overlap of their opening of 10 mm. Riser tube has OD/ID of 9.52 mm/ 8.3 mm and 300 mm long. While the downcomer tube has ID of 5 mm and 225 mm long. The manufactured TP is shown in figure 6.

Figure 6. Manufactured TP

The vacuum gauge is calibrated according to a standard calibration procedure with a pre-calibrated vacuum gauge. The flow meter is calibrated according to a standard calibration procedure with graduated cylinder and stop watch.

Uncertainty analysis is performed for the instrumentation used in the experimental set–up. For each measured parameter, one or more elemental uncertainties can contribute to the total uncertainty for the parameter. These elemental uncertainties are combined using the uncertainty method by (A. Wheeler...
et.al., 2010) [39]. The total uncertainty in measuring the heater input power, average evaporator-condenser temperature difference and $R_{th}$ are in the range (0.83–3.2) % depending on the test settings. The testing conditions are: FR (40, 50, 55, 60, 70) % of the evaporator section volume. Where the FR is the ratio of the working fluid volume to the evaporator section volume. The supplied heat (50, 75, 100, 150, 200) Watt. The inclination angle (15°, 30°, 45°, 60°, 75°, 90°) with respect to horizontal plane.

3. Thermal Performance

There are different ways used to summarize the thermal performance of THP for competition purpose between different configurations. These different configurations could be working fluid, pipe material, size/geometry and filling ratio. The competition can be done between different working parameters or boundary conditions such as the supplied heat, cooling system and inclination angle [40][2][41]. The most favorable equation used for competition purpose is the thermal resistance ($R_{th}$) equation [2], [3], [42], [43]:

$$R_{th} = \frac{T_{evap.avg.} - T_{cond.avg.}}{Q_{input}}$$

(1)

Therefore, the lower thermal resistance value for the THP is the better thermal performance among all suggested and tested configurations for the same boundary/operating conditions.

The enhancement percentage of the thermal performance is calculated from the reduction percentage for the $R_{th}$ and calculated as follows:

$$\% \text{ Reduction of } R_{th} = \frac{R_{th,normal} - R_{th,enhanced}}{R_{th,normal}}$$

(2)

4. Results and Discussion

Samples of the experimental results for temperatures distribution along the THP and the TPTHP are plotted together in figures 7 a-c. At the evaporator section and at the adiabatic section, the TPTHP temperatures trend lines are lower than that for THP at the same operating conditions. While at the condenser section the trends seem to be together.
The verification of the thermal performance enhancement due to the use of the TP to control the flow paths of the two-phases of the THP is the main goal for this study. As proposed earlier, the insertion of the TP regulate / control the flow paths of the streams of the two phases. Then enhance the evaporation and condensation processes. The temperature distribution for the TPTHP show a relatively lower temperatures distribution values at evaporator and adiabatic sections w.r.t. THP. These effects will lead to enhance the thermal performance of the operation of the THP. This enhancement could be found/calculated from the reduction of $R_{th}$ values. Samples for the experimental results of $R_{th}$ values for THP and TPTHP are plotted in figures 8 a-c for comparison purpose. The figures showed that the insertion of TP reduces $R_{th}$ values with noticeable magnitudes relative to that of normal geometry. The curves of $R_{th}$ for TPTHP are seemed as identical despite of different operating conditions. The variation between them is very small and sometimes it seems as superimposed on each other (the effects of FR and inclination angle are9 discussed later). All trends of $R_{th}$ for the THP and TPTHP show a decrease in $R_{th}$ values related to the supplied heat. Also $R_{th}$ decreasing slop of the TPTHP w.r.t the supplied heat, are lower than that for the THP.

Figure 7,a-c Experimental Temperature distribution along THP and TPTHP for a Sample Cases
Figure 8a-c. Experimental Results for the Effect of TP Insertion on Thermal Resistance Values of THP
The thermal performance enhancement percentages or reduction of $R_{th}$ percentages are calculated using eqn. (1) and three samples are plotted in figures 9 a-c. The plots showed that the reduction percentages of $R_{th}$ w.r.t. the supplied heat is not constant and varies from one case to another. Also there is no a clear behavior for this relation. Thus there are other parameters that controls the relationship between the reduction percentages of $R_{th}$ and the supplied heat. These parameters are the FR and the inclination angle.

Maximum enhancement percentages for the thermal performance is found at the maximum reduction of $R_{th}$ value from the experimentally tested parameters was about 62%. This $R_{th}$ reduction percentage was for operating conditions of FR 40%, inclination angle: 15º and heat supplied: 50 W. While, a minimum enhancement percentage for the thermal performance is found at the minimum reduction of $R_{th}$ value from the experimentally tested parameters was about 17 %. This $R_{th}$ reduction percentage was for operating conditions of FR 55%, inclination angle: 60º and heat supplied:100 W.

**Figure 9 a-c.:** Reduction Percentages of Thermal Resistance between THP and TPTHP
To study the effect of the insertion of the TP on the transient operation for the THP, the average evaporator section temperatures, adiabatic section temperatures and the averaged condenser section temperatures behaviors figure 10 is plotted for operating conditions of FR: 50%, Incl. 45° and supplied heat of 200W. The figure showed that the evaporator section temperatures for the TPTHP has a lower averaged temperature ramp at starting with respect to the THP operation. This is may be attributed to the faster regulation for evaporation and condensation processes. The generated steam in the evaporator section forwarded directly to the condenser section through the riser tube. The condensate returns to the evaporator section through the downcomer tube. This evaporation/condensation cycle starting/regulating in the TPTHP faster than that for the THP. The insertion of TP reduces evaporation section maximum averaged temperature during transient operation about 50% for the tested operating conditions. This reduction for the maximum evaporator section temperatures is a good advantage for the using of the TP. Because it reduces the risk of using of the THP in cooling of electronic circuits/chips. Where these components temperatures should not exceed 80 °C [8], [44]. After the temperature ramp, the evaporator section temperatures go smoothly toward steady state temperature.

The adiabatic section temperature trend shown as uniform/regular trend start from the start-up temperature and go towards the steady state temperature without any ramp.

The condenser section temperature trend shown as uniform/regular trend start from the start-up temperature and go towards the steady state temperature.

The insertion of the TP affect the period required to reach steady state operation. Because it regulates the evaporation and condensation processes cycle faster than that for the THP. The steady state operation for the TPTHP is achieved within about (10-12) minutes. This period was less than that of the THP steady state period which is about (16-19) minutes. Thus, the insertion of TP reduces the transient operation time about 37% for the tested operating conditions.

**Figure 10.** A Sample of Experimental Transient operation Temperatures Behaviour for the THP and the TPTHP

Figure 11 showed a sample for the effects of FR on the THP thermal performance. The $R_{th}$ trends clearly appeared as separated curves and varied due to the variations of FR. The TPTHP trends for $R_{th}$ appeared as coincided/superimposed trends. There are not clear trends for $R_{th}$ due to variation of FR.
These coinciding curves show that the thermal performance of TPTHP operation is not/slightly influenced due to variation of FR. This is because the insertion of TP controls the flow paths of the generated steam and the condensate streams. The working fluid circulate smoothly from the evaporation section to the condensation section. Then returning to the evaporation section during the evaporation and condensation processes. The working fluid filling the fluid domain regularly without obstructions of complex circulation fields. These fields consume some of the working fluid volume and/or prevent the smooth circulation of the working fluid during evaporation and condensation processes cycle. Finally, the addition of extra amounts of working fluid (from 40% to 70% of the evaporator section volume) will follow the smooth circulation and not built/lead to flooding.

The slope of decreasing $R_{th}$ values due to the increase of the heat supply for the TPTHP operation, was relatively less than that for the THP operation.

Figure 12 showed a sample for the optimization curves for the effect of FR on $R_{th}$ values for the TPTHP. There is no clear trend/behavior for the effect of FR on $R_{th}$. The $R_{th}$ values fluctuated within a narrow range of values for each test conditions of inclination angles and supplied heats. The insertion of TP reducing/vanishing the effect of FR on $R_{th}$ values. Thus, the thermal performance of the TPTHP is not affected by the variations of FR. Then the TPTHP can operate at relatively low values of FR (40%) with the same thermal performance for higher values. The required amount of the working fluid for the enhanced thermal performance of the TPTHP was (FR 40%). This amount was lower than that for the THP which was (FR 55%) [34].

The reduction percentages for $R_{th}$ values due to variation of FR between the THP and the TPTHP at different testing conditions are plotted in figure 13. The curves showed that the highest reduction percentages curves are for FR 40%. The insertion of TP reduces the required FR for optimum operation. The high reduction percentage of $R_{th}$ value for the TPTHP at FR 40% are due to the high value of $R_{th}$ at FR 40% for the THP operation. Hence the high difference of $R_{th}$ values before and after the insertion of TP will produce relatively high reduction percentages of $R_{th}$ values.

![Figure 11. Sample for the Effect of TP on Thermal Resistance at various Operating Conditions of THP and TPTHP](image-url)
To study/discuss the effects of the inclination angles with various operation conditions of FR and heat supplied on the TPTHP thermal performance or $R_{th}$. A sample of the experimental tests for the THP and the TPTHP are plotted in figure 14. The figure showed the effect of the inclination angles on the THP thermal performance/$R_{th}$ is clear appeared as separated curves. The $R_{th}$ trend varies due to the variations of the inclination angles for the THP. While the TPTHP trends for $R_{th}$ appeared as coincided/superimposed trends. There are not clear trends for $R_{th}$ due to variation of the inclination angles. These coinciding curves showed that the thermal performance of TPTHP operation are not/slightly influenced due to variation of the inclination angles. Because the insertion of the TP control the flow paths of the generated steam and the condensate streams. The working fluid circulate smoothly from the evaporation section to the condensation section. Then returning to the evaporation section during the evaporation and condensation processes. Provide a relatively good distribution for the steam in the condenser section around the riser tube at all inclination angles. The decreasing slop of $R_{th}$ values due to the increasing of the heat supply for the TPTHP operation was less than that for the THP operation.

Figure 15 showed a sample for the optimization curves for the effect of the inclination angles on $R_{th}$ values for the TPTHP at various FR and supplied heat. There is no clear trend/behavior for the effect of the inclination angles on $R_{th}$. The $R_{th}$ values fluctuated within a narrow range of values for each test.
conditions of FR and supplied heats. So that the insertion of TP reducing/vanishing the effect of the inclination angles on $R_{th}$ values. The thermal performance of the TPTHP is not affected by the variations of the inclination angles. Hence the TPTHP can operate at different inclination angles with the semi-constant thermal performance.

Sample for the reduction percentages for $R_{th}$ values due to variation of inclination angles between THP and TPTHP experimental operations are plotted in figure 16. The curves showed that the highest reduction percentages curves are for the inclination angle of 15°. This relatively high reduction percentage for the inclination angle of 15° for the TPTHP are due to the lowest values of $R_{th}$ for the inclination angle 15° for the THP operation. Thus the high difference of $R_{th}$ values before and after the insertion of TP will produce relatively high reduction percentages of $R_{th}$ values.

**Figure 14.** Sample for the Experimental Thermal Resistance of THP and TPTHP at Various Inclination Angles

**Figure 15.** Sample for the Optimization of the Effect of The Inclination Angles on Thermal Resistance for TPTHP
5. Conclusions
A two phase separation technique is tested experimentally to verify the enhancement of the thermal performance of the THP using tubes packing (TP). The experimental results are discussed and leading to the following conclusions:

a- The used separation technique is proved to work well. So that it enhanced the thermal performance of the THP by reducing the transient operation time by about 37%.
b- The maximum averaged temperature of the evaporator section are reduced by 50%.
c- The effect of filling ratios and inclination angle on the thermal performance of the THP is reduced/dismissed when the TP insertion is used.
d- The thermal resistance of the THP is noticed with a reduction of 17% to 62% based on the operating condition when using the TP insertion.

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Abbreviations

| Abbreviation | Description                        |
|--------------|-----------------------------------|
| DIW          | De-Ionized Water                  |
| FR           | Filling Ratio                     |
| HP           | Heat Pipe                         |
| HVAC         | Heat Ventilation and Air Conditioning |
| SVF          | Steam Volume Fraction             |
| VF           | Volume Fraction                   |
| VOF          | Volume Of Fluid                   |
| THP          | Thermosyphon Heat Pipe            |
| TP           | Tubes Packing                     |
| TPTHP        | Thermosyphon Heat Pipe with Tubes Packing |
Nomenclature

| Symbol | Description | Unit |
|--------|-------------|------|
| ID/OD  | Inside / Outside diameter | m    |
| R_th   | Thermal resistance | K/W  |
| T      | Temperature | K    |
| t      | Time | Sec  |
| t      | Thickness | m    |
| T_evap.avg. | Average temperature of evaporator section | K |
| T_cond.avg. | Average temperature of condenser section | K |
| Q      | Supplied heat | Watt |

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