Numerical Simulation of the Heat Transfer Characteristics of Low-Watt Thermosyphon Influence Factors

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Abstract

This study used numerical simulation to analyze the influence of different factors, such as vacuum degree, fill ratio and aspect ratio, on the heat transfer characteristics of low-watt thermosyphon in natural convection. The mass and energy source terms were added in the continuity and energy equation to simulate the exchanges between vapor and liquid phases. The comparison analysis of computed result and experiment data showed that the numerical model used in this study is appropriate to analyze the physical mechanism and heat transfer property of thermosyphon. The results confirmed that low vacuum degree and long evaporator characterize better thermal resistance. The optimal fill ratio was 60% when the vacuum degree was 35 torr and the aspect ratio was 11.8.

Key Words: Heat Pipe, Thermosyphon, Vacuum Degree, Fill Ratio, Aspect Ratio, Boiling Heat Transfer

1. Introduction

The excessive consumption of fossil energy has led to energy poverty in recent years; hence, the development of alternative energy has become a widely concerned issue. Among the current alternative energy sources, solar power generation [1] and geothermal power generation are popular and widely adopted. However, considering the high equipment costs and limited power generation efficiency, thermosyphon, which is characterized by simple structure [2], low cost, and high efficiency, has become predominant. Due to high thermal conductivity, the thermosyphon has been extensively used for frozen soil conservation [3], snow thawing, and tunnel frost prevention. The thermosyphon was first proposed by E. Schmidt [4] in 1951, who suggested that even if the temperature difference between evaporation zone and condensation zone was small, the heat transfer was still efficient. There were many subsequent studies on the heat transfer efficiency and application of thermosyphon. In 1983, Imura et al. [5] used water, ethanol and trichlorotrifluoroethane as working fluid to correct experimental data, and established the critical heat flux equation of closed two-phase thermosyphon. They found that the internal pipe diameter, heating range, working fluid, loading and vacuum degree of thermosyphon were important factors influencing the critical heat flux. In 2005, Noie [6] conducted experiments with three factors that influence the vertical closed two-phase thermosyphon, namely input heat transfer capacity, working fluid fill ratio, and evaporation end length ratio (A.R.: aspect ratio is the ratio of evaporation end length to internal diameter of pipe). The deionized water was used as working fluid. The experimental results showed

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that in different evaporation end lengths, the maximum heat transfer capacity occurred under different fill ratios. The highest heat transfer capacity of evaporation end length ratio of 11.8 occurred at the fill ratio of 60%. The highest heat transfer capacity of evaporation end length ratios of 7.45 and 9.8 occurred at the fill ratios of 90% and 30%, respectively. In 2013, Kafeel et al. [7] used a fill ratio of 30% and Eulerian model of ANSYS/Fluent 13.0 to simulate the two-phase variation of thermosyphon with different relaxation time values and input heat fluxes. The simulation results were very close to the experimental results. Although the accuracy of Eulerian model was better than that of the Mixture model, the computing and stability was worse than that of the Mixture model [8]. Therefore, this study adopted the Mixture model.

Due to the excessive consumption of energy and the occurrence of energy crisis, the thermosyphon has been used extensively. However, most of current research data are experimental data, while there are few numerical simulations on two-phase changes in the tube. Therefore, this study used the Fluent 14.5 of ANSYS software to numerically simulate the heat transfer mechanism and physical phenomenon. Three factors influencing the thermosyphon efficiency were discussed, including four vacuum degrees (35, 55, 65, 75 torr), three fill ratios, namely the volume ratio of liquid (F.R. = 30%, 60%, 90%), and three aspect ratios (A.R. = 7.45, 9.8, 11.8). The working fluid was pure water. The three factors were used to obtain the optimal application conditions of thermosyphon as the base of future development of thermosyphon.

2. Research Method

2.1 Numerical Simulation

The experimental model developed by Noie [6] is shown in Figure 1. The overall length was 980 mm; the outside diameter was 32 mm; the wall thickness was 3.5 mm; the condensation section length was fixed at 380 mm; the aspect ratio (A.R.) was defined as evaporator length/inside diameter of 7.45 (evaporator length 238 mm; insulation section length 362 mm), 9.8 (evaporator length 314 mm; insulation section length 286 mm) and 11.8 (evaporator length 377 mm; insulation section length 223 mm) respectively.

2.2 Assumed Conditions

The assumptions for simulation include:

(1) Steady-state heat transfer.
(2) The fluid flow field inside heat pipe is laminar flow.
(3) The vapor is saturated vapor.
(4) Under gravity effect.
(5) The liquid and vapor fluid are Newtonian fluids.
(6) Maintain the working fluid circulation, there is no dry-out.

2.3 Governing Equation

Based on the experimental research on thermosyphon, this study adopted the Mixture model of commercial kit Fluent 14.5 for numerical simulation of thermosyphon. Source term was added in the mass and energy equations to simulate the vapor and liquid changes in the working fluid.

Continuation equation of Mixture model:

\[
\frac{\partial}{\partial t} (\rho_m) + \nabla \cdot (\rho_m \vec{v}_m) = \dot{m}
\]

(1)

where \( \vec{v}_m \) is the mass average velocity

\[
\vec{v}_m = \frac{\sum \alpha_k \rho_k \vec{v}_k}{\rho_m}
\]

(2)

\( \rho_m \) is the mixture density

\[\rho_m = \sum_{k=1}^{n} \alpha_k \rho_k\]

(3)

\( \alpha_k \) is the volume fraction of the \( k \)-th phase.

\( \dot{m} \) is the mass transfer of source term in the mass equation of user defined function, assuming that the mass transfer process during evaporation and condensa-

![Figure 1](image-url)
tion of working fluid is steady-state process. The vapor-liquid phase change is determined by the saturation temperature. Source terms of liquid phase to vapor phase and vapor phase to liquid phase are \[9,10\].

\[
\dot{m}_l = \begin{cases} 
\frac{r_l \alpha_l (T_l - T_{sat})}{T_{sat}} & T_l \geq T_{sat} \\
0 & T_l < T_{sat}
\end{cases}
\]

\[
\dot{m}_v = \begin{cases} 
\frac{r_v \alpha_v (T_v - T_{sat})}{T_{sat}} & T_v \leq T_{sat} \\
0 & T_v > T_{sat}
\end{cases}
\]

where \(r_{vl}\) and \(r_{lv}\) are the time relaxation factors. \(\alpha_v\) and \(\alpha_l\) are the vapor-liquid volume ratio function, and \(\alpha_v + \alpha_l = 1\). \(\rho_v\) and \(\rho_l\) are the vapor-liquid density ratio function. \(T_{sat}\) is the saturation temperature, \(T_l\) is the liquid phase temperature, \(T_v\) is the vapor phase temperature.

Momentum equation of Mixture model:

\[
\frac{\partial}{\partial t} (\rho_m \vec{u}_m) + \nabla \cdot (\rho_m \vec{u}_m \vec{u}_m) = -\nabla p + \nabla \left[ \mu_m (\nabla \vec{u}_m + \nabla \vec{u}_m^T) \right] + \rho_m \hat{g} + F + \nabla \sum_{k=1}^{n} \alpha_k \rho_k \vec{V}_{k,k} \vec{V}_{v,k}
\]

where \(n\) is the number of phases, \(F\) is the body force, \(\mu_m\) is the mixture viscosity

\[
\mu_m = \sum_{k=1}^{n} \alpha_k \mu_k
\]

\[
\vec{v}_{v,k} = \vec{v}_k - \vec{v}_m
\]

where \(\vec{v}_{v,k}\) is the slip velocity of the second phase \(k\).

Energy equation of Mixture model:

\[
\frac{\partial}{\partial t} \sum_{k=1}^{n} (\alpha_k \rho_k E_k) + \nabla \cdot \sum_{k=1}^{n} (\alpha_k \vec{v}_k (\rho_k E_k + p)) = \nabla \cdot (k_{eff} \nabla T) + S_E
\]

\(k_{eff}\) is the effective thermal conductivity, \(S_E\) contains other volumetric heat sources

\[
E_k = h_k - \frac{P}{\rho_k} + \frac{v_k^2}{2}
\]

The compressible phase \(E_k\) is Eq. (10), the incompressible phase \(E_k = h_k\), \(h_k\) is the sensible enthalpy of the \(k\)th phase.

### 2.4 Initial and Boundary Conditions

The fill ratio of the working fluid in this paper is 30, 60 and 90% respectively. The evaporator heat input is \(Q_w = k_w A_w \frac{dT}{dx} = 40W\); the condensation section cooling temperature is 298 K (25 °C); the gravitational acceleration of working fluid returning from condensation section to evaporator is 9.81 ms\(^{-2}\).

The boundary conditions are set as follows:

1. Condensation section: convection heat transfer \(q_w = h(T_w - T_{w0})\)
2. Insulation section: zero heat flux \((\frac{\partial T}{\partial x})_w = 0\)
3. Evaporator: fixed heat input \(q_w = -(k \frac{\partial T}{\partial x})_w\)

### 2.5 Definition of Thermal Resistance

\[
R_{th} = \frac{T_{w,ave} - T_{c,ave}}{Q}
\]

where \(T_{w,ave}\) is the average temperature of heating wall surface, \(T_{c,ave}\) is the average temperature of condensation wall surface, \(Q\) is the heating capacity.

### 2.6 Numerical Modeling

The Mixture model of Fluent 14.5 two-phase flow module [8] was used. The primary phase is the liquid phase and the secondary phase is the vapor phase. The pressure-velocity coupling uses SIMPLE algorithm, and the discrete momentum and volume ratio equations use QUICK method. The energy equation is solved by the first order upwind scheme. The solver is unsteady implicit iteration with time step equal to 0.01 sec. The accuracy of the numerical computation is \(10^{-4}\).

### 3. Results and Discussion

This section discusses the influence of the three factors of thermosyphon on the heat transfer characteristics. The thermosyphon introduced by M. Karthikeyan [11] was used for simulation. In order to ensure the heat transfer accuracy of the thermosyphon, the proximal edge encrypted rectangular grid was used, and the total number of grids of overall computational domain was about 30,000. The boiling point of the working fluid was 350 K
Therefore, when the saturation temperature of liquid working fluid exceeded 350 K (77 °C), the working fluid evaporated; on the contrary, when the saturation temperature of vapor working fluid was lower than 350 K, the condensation occurred. As shown in Figure 2, the internal fill ratio of thermosyphon was set as 60% for computing. The simulation results were close to the experimental result.

The model developed by Noie [6] was used to compare different vacuum degrees (35, 55, 65 and 75 torr). The simulation results are shown in Table 1. As seen, the saturation temperature of the internal working fluid decreased with the pressure. When the saturation temperature was low, the thermosyphon started up earlier. Therefore, when the vacuum degree was low, the thermal resistance was also low. The results are shown in Figure 3. The thermal resistance value was better when the va-

<table>
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<th>Vacuum (torr)</th>
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<th>Aspect ratio</th>
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<th>Top temp. (K)</th>
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The vacuum degree was 35 torr.

At the fill ratio of 60% and different aspect ratios A.R. = 7.45, 9.8, 11.8, when the heating zone was long, the working fluid reached saturation temperature quickly, so that the syphon was started up. The results are shown in Figure 4. Therefore, the thermal resistance value was better when A.R. = 11.8.

Figure 5 shows the results when the fill ratios were 30, 60 and 90%, the vacuum degrees were 35, 55, 65 and 75 torr, and A.R. = 11.8. The optimum thermal resistance was obtained when the fill ratio was 60%, which is consistent with the results of Noie [6].

4. Conclusions

(1) The comparison of the experimental results proved that the simulation is effective on the heat transfer mechanism and physical phenomenon of thermosyphon.

(2) The results indicated that at a low vacuum degree, the thermal resistance was low, and the thermal conductivity of thermosyphon was good.

(3) The thermal resistance decreased obviously as the aspect ratio increased, the lowest thermal conductivity was obtained when A.R. = 11.8.

(4) When the vacuum degree was 35 torr and A.R. = 11.8, the fill ratio of 60% resulted in the optimal thermal resistance.

According to the above results, this study confirmed the simulation and optimization of the characteristics of thermosyphon to be feasible. Our future work will focus on high-watt thermosyphon.

References


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