Applied Thermal Engineering 108 (2016) 878–892

Contents lists available at ScienceDirect

Applied Thermal Engineering

journal homepage: www.elsevier.com/locate/apthermeng

Research Paper

Improvement of a novel heat pipe network designed for latent heat thermal energy storage systems

Mahboobe Mahdavi, Songgang Qiu *, Saeed Tiari

Department of Mechanical Engineering, Temple University, 1947 N. 12th Street, Philadelphia, PA 19122, USA

HIGHLIGHTS

- Investigation of the effects of heat pipe network configuration on its performance.
- Modeling of combined vapor jet impingement and vapor condensation.
- Optimized geometry to alleviate flow separation and improve the performance.
- Study of the effects of condenser inlet shapes and adiabatic section locations.
- Best performance was achieved with concentric adiabatic section configuration.

ARTICLE INFO

Article history:
Received 18 March 2016
Revised 24 July 2016
Accepted 28 July 2016
Available online 29 July 2016

Keywords:
Heat pipe network
Primary heat pipe
Secondary heat pipe
Jet impingement
Numerical simulation
Latent heat thermal energy storage systems

ABSTRACT

In the present work, the performance of a heat pipe network designed for a latent Thermal Energy Storage unit in a Concentrating Solar Power System was investigated numerically. A two-dimensional axisymmetric model was implemented to describe the vapor flow and heat transfer inside the vapor core. The network consists of a primary heat pipe and a concentric secondary heat pipe. The solar energy impinges on the disk shaped evaporator is transferred to the heat engine through adiabatic section. The excess heat is used to charge the phase change material via the secondary concentric heat pipe. The vapor flow leaving the adiabatic part of the primary heat pipe to the main condenser is similar to the confined jet impingement. As the flow impinges on the surface and spreads out radially, several recirculation zones have formed, resulting in non-uniform condensation on the condenser surface. The objective of the current work is to optimize the geometry of the heat pipe to alleviate flow separation and hence to improve the performance of the heat pipe. The effects of main condenser and secondary heat pipe entrance shapes on streamline contours, pressure and temperature distributions of the main condenser and secondary heat pipes were investigated. The impact of the primary heat pipe position was studied. Two configurations studied are center located adiabatic section and outward positioned adiabatic section. The performance of the heat pipe was evaluated by calculating the corresponding thermal resistances. For a case with tubular adiabatic section, the result showed that the condensers inlets shapes do not have significant effects on the recirculation zone configuration, but only shift up the pressure and temperature distributions of the main condenser and the secondary heat pipe. Among the various shapes studied for the main condenser inlet, tapered inlet results in the lowest thermal resistances for both the main and secondary condensers. It was also concluded that locating the adiabatic section of the primary heat pipe outwards reduces the size and quantity of the recirculation zones and dramatically increases the average temperatures of the condensers. Tapering the main condenser entrance eliminates the primary recirculation zone, resulting in a uniform temperature distribution inside the main condenser.

1. Introduction

Heat pipes have been proven to be one of the most efficient heat transfer devices. They have simple and reliable operation with no moving parts [1]. Heat pipes have the ability to transfer high rate of heat with a small temperature difference over their length. In heat pipes, heat transfer occurs through the evaporation and condensation of a working fluid enclosed in a vacuumed and sealed container. The applied heat to one end of the heat pipe named evaporator changes the phase of the working from liquid to vapor. The generated vapor travels to the other end of heat pipe called

---

* Corresponding author.
E-mail address: songgang.qiu@temple.edu (S. Qiu).

http://dx.doi.org/10.1016/j.applthermaleng.2016.07.190
1359-4311/

Nomenclature

c_p	specific heat, J/kg-K
h_{fg}	latent heat of evaporation, J/kg
k	thermal conductivity, W/m-K
L	length, m
n	normal direction
P	relative pressure ($P_{absolute} - P_{refrence}$), Pa
Pct_q	heat transfer via secondary heat pipe in percentage of total heat input to the evaporator.
P_o	reference pressure, Pa
q''	heat flux, W/m^2
Q	heat transfer, W
r	radial coordinate
R	radius, m
R_i	inlet corner radius, m
R_g	gas constant, J/kg-K
R_v	adiabatic section radius
S	secondary heat pipe spacing
T	temperature, K
T_o	reference temperature, K
t	thickness, m
t_v	adiabatic section thickness, m
u	axial velocity, m/s
v	radial velocity, m/s
z	axial coordinate

Greek

ρ	density, kg/m^3
μ	dynamic viscosity, N s/m^2
φ	viscous dissipation

Subscripts

c	condenser
e	evaporator
in	input
Int	interface
l	lower
liq	liquid
t	top
o	outlet
v	vapor
w	wall
i	inlet
s,i	secondary inlet

condenser where it condenses and releases its heat to heat sink. The inner surface of the heat pipe container is covered with a wick material, which provides the capillary force to return the liquid back from condenser to evaporator.

Various numerical investigations have been performed and reported on the hydrodynamics and thermal performance of conventional heat pipes. The simplified models only dealt with steady state vapor flow dynamics in conventional cylindrical [2] or flat heat pipes [3]. The heat pipe simulations were extended to include the liquid flow in the wick [4], conjugate heat transfer effects [5] and coupling of vapor and liquid phases [6]. Two general techniques are often used to calculate evaporation and condensation rates at the wick-vapor interfaces. In the first approach, evaporation and condensation rates are calculated using energy balance equation at the cells adjacent to the interface knowing the interfacial heat flux [7]. Using this method, Koito et al. formulated a mathematical model to simulate the thermal-fluid phenomena in a copper-water heat pipe. Calusius-Clapeyron equation was implemented to calculate the saturated temperature [8]. In the second approach, kinetic theory of gasses is used to estimate the evaporation and condensation rates. In this method, the saturation temperature is obtained from the energy-balance equation at the interface. The saturation pressure is calculated from Calusius-Clapeyron as a function of saturated pressure [9]. The operations of heat pipe can also be described using thermal resistance network methods. Using this technique, the overall thermal resistance of the heat pipe is obtained based on the thermal resistances associated with each individual component and the process of heat pipes [10].

Due to their design flexibility, heat pipes can be used in many applications such as electronic cooling [11], waste heat recovery [12] or in aerospace applications to cool the leading edges of reentry vehicles and hypersonic aircrafts [13]. Recently, there has been a flourishing interest in exploring the utilization of heat pipes in latent heat thermal energy storage systems for Concentrating Solar Power (CSP) generation application.

In Concentrating Solar Power applications, Sun's energy is collected via the utilization of mirrors and a concentrator. The concentrated energy is then employed to run a heat engine for electricity production. Latent heat thermal energy storage systems combined with CSP systems alleviates the intermittency issue of the solar energy sources and reduces the cost of energy.

One advantage of using latent heat energy storage systems is the large amount of energy associates with the heat of fusion resulted from solid-liquid phase change that tremendously reduces the size and cost of the system [14]. Another advantage of latent heat thermal energy storage systems is nearly isothermal operation of the heat engine. The materials that are used in latent heat thermal energy storage systems to store energy are referred as phase change materials (PCM) [15].

Despite the advantages mentioned above, the performance of latent heat thermal energy storage systems is often hindered by low thermal conductivity of commonly used, low cost phase change materials (PCMs). This low thermal conductivity results in high thermal resistance between the heat source and the PCM and leads to prolonged charging and discharging processes of the systems, as well as overheating the heat transfer surface. Research and development of passive heat transfer devices, such as heat pipes (HPs) to enhance the heat transfer in the PCM has received considerable attention [16].

The focus of the majority of previous studies was to improve the performance of the latent heat thermal energy storage system by implementing different arrangement of conventional cylindrical heat pipes.

Sharifi et al. explored the effect of a single cylindrical heat pipe on the melting of a phase change material in a vertical cylindrical container. The container was heated by the heat pipe, which was located at the center and its condenser was surrounded by the PCM. The performance of the heat pipe integrated system was assessed by comparing the melting fraction at each time to those of assisted by an isothermal surface or a hot concentric rod or tube. Higher melting rates were reported for the case with heat pipe [17].

Motahar and Khodabandeh studied the effect of a heat pipe on both charging and discharging processes of a phase change material enclosed by a cylindrical container. They also reported dramatic improvements in the PCM melting and freezing rates [18]. Robak et al. assessed the effectiveness of latent heat thermal