International Communications in Heat and Mass Transfer 39 (2012) 494–503

Contents lists available at SciVerse ScienceDirect

International Communications in Heat and Mass Transfer

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

Dynamic test strategy for diagnosing a heat pipe cooling module ☆

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ARTICLE INFO

Available online 21 February 2012

Keywords:
Heat pipe cooling module
Steady-state test
Dynamic test

ABSTRACT

This article experimentally develops a dynamic test strategy for efficiently diagnosing a heat pipe cooling module in order to improve the time-consuming conventional steady-state test. The first step is to investigate the performance of a heat pipe by measuring its thermal resistance, and the next step is to examine the influence of the parameters on the temperature response of the heat pipe cooling module. The experimental parameters include the press force, preheating temperature, heating power, and starting time of the fan. The results show that the thermal performance of a heat pipe, the contact condition between the heat pipe and the base plate, and the heat dissipation ability of a heat sink, are diagnosed within 30 seconds. During the dynamic test, both the startup and the ability to reach uniformity of temperature of the heat pipe can be observed. In addition, the temperature response of a heat pipe cooling module based on a lumped model matches the experimental data.

1. Introduction

The investigation of heat pipes and their applications in thermal management have been known for years. However, they have now become more attractive for transporting heat in electronics such as notebook PCs, which use a heat pipe applied to a cooling module to remove heat. A cooling module is composed of a heat pipe, a heat sink with fins, and a fan. The heat generated from a localized heat source is absorbed at the evaporation section of the heat pipe and is then transported to the condensation section to be released into the heat sink. Eventually, the heat is removed and dissipated by either free or forced convection. To ascertain the heat dissipation function of a heat pipe cooling module, its performance should be tested after manufacture. The test includes the contact condition between each component of a heat pipe cooling module in order to observe heat conduction. Through the thermal resistance model, the contact condition between each component of a heat pipe cooling module can be analyzed with measured temperature and heating power [1–10].

Furthermore, it is necessary to inspect the heat pipe in depth before examining the cooling module, as the heat pipe plays an important role in heat transportation. There were many experiments and simulation models [11–16] in investigating the heat pipe during the past decade, including the measurement of maximum heat transfer rate and working fluid filled ratio, the inside surface structure, the experimental and analytical investigation of operating parameter effects, and the investigation of transport phenomena inside heat pipes.

Several works were recently carried out to study transient thermal performances. The concept of heat pipe time constants is introduced to describe the transient behavior of heat pipes, which can promote understanding of the startup and shutdown phenomena of heat pipes [17,18]. Tsai et al. [19] proposed a dynamic test that shortens the necessary time for determining the thermal performance of heat pipes. The parameter is defined as "decreasing slope of temperature difference $°Cs^{-1}$", which effectively reflects the parameter of the steady-state test. Murer et al. [20] developed a one-dimensional model of a copper–water miniature heat pipe, which can be applied to both steady-state and transient simulations and enables the determination of the time evolution of the temperature along the heat pipe for arbitrary distributions of heat sources and sinks. Faghri et al. [21,22] adopted a lumped model to simulate the temperature response of a heat pipe. This model can predict the temperature response of a heat pipe and the time needed to reach steady-state time. The analytical solutions for the full two-dimension conservation equations and experimental data were found to be nearly identical.

Dobre et al. [23] applied cooling systems based on the heat pipe principle to control the operating temperature of electronic components. He proposed an experimental setup and a data processing algorithm that can be used to establish the heat transfer characteristics of an electronic device cooler based on heat pipes fitted with a fins system. However, the test methodology requires a long time to characterize the performance of cooling modules; therefore, it is not appropriate to apply this test method to industrial production.

The objective of this study is to propose a strategy for efficiently diagnosing the heat pipe cooling module. The strategy is to examine

☆ Communicated by W.J. Minkowycz.

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doi:10.1016/j.icheatmasstransfer.2012.02.007

Nomenclature

A	area (m^2)
C_p	specific heat (kJ kg^{-1} °C^{-1})
h	heat transfer coefficient (W m^{-2} °C^{-1})
k	thermal conductivity (W m^{-1} °C^{-1})
K	permeability of wick structure (m^2)
M	mass (kg)
$\dot{Q}$	heat transfer rate (W s^{-1})
Q	input power (W)
R	thermal resistance (°C W^{-1})
S	decreasing slope of temperature difference (°C s^{-1})
T	temperature (°C)
U_s	internal energy (kJ)
u	specific internal energy (kJ kg^{-1})
v	specific volume (m^3 kg^{-1})
X	quality

Greek symbol

ΔT	temperature difference between evaporation and condensation sections (°C)
ΔT_{ca}	temperature difference between condensation section and ambient (°C)
δ	thickness of the wick structure (m)
ε	wick porosity

Subscripts

a	ambient
c	condensation
e	evaporation
eff	effective
i	input
l	liquid
o	external
s	saturation
v	vapor
w	wick
max	maximum

the thermal performance of a heat pipe, the cooling performance of forced convection, and the thermal resistance between devices, including contact resistance and interface resistance. The study adopts constant heat flux as a fixed heat source to observe the temperature responses of the cooling module considering press force, preheating temperature, heating power, and starting time of the fan. The experimental results are discussed in order to determine criteria for developing the test strategy.

2. Experimental procedure and setups

In this study, the experimental procedures consist of steady-state and dynamic tests. Fig. 1 demonstrates that the experimental setup of the steady-state test, which measures the thermal contact resistance between the base plate and the heater, as well as the Q_{max} of the heat pipe, through thermal resistance analysis. The dynamic test develops a strategy for diagnosing a heat pipe cooling module, as shown in Fig. 2. Detailed descriptions are provided in the following sections.

2.1. Steady-state test

The conventional test adds thermal grease to the surface between the base plate and the heater to decrease contact thermal resistance. However, this procedure increases costs and test time. In addition, the Q_{max} of the heat pipe is examined before starting the dynamic test. In this study, we use thermal resistance analysis to measure thermal contact resistance and Q_{max}.

Fig. 1a schematically presents the test facility, which includes a power supply, a data recorder, and a strain gauge. The evaporation of the heat pipe uses a dummy heater to simulate electronic device heat dissipation. The heater, powered by a DC power supply, is finely insulated with Plexiglas and Styrofoam. The base plate is placed on the top of the heater—the contact area between the heater and the base plate is 15×15 mm^2—then both are put on the load cell. The press force from the base plate to the heater is measured with the strain gauge. After heating the base plate, the temperature responses are recorded with a Yokogawa data recorder.

2.1.1. Thermal resistance analysis

The base plate of the cooling module is placed on top of the heater and they are then stamped together. The thermal resistance network of the cooling system is presented in Fig. 1b. The total thermal resistance is the sum of contact resistance R_c, spreading resistance R_{sp}, conduction resistance R_m, and interface resistance $R_{b\text{-}e}$, expressed as

$$R_e = \frac{T_h - T_e}{Q} = R_c + R_{sp} + R_m + R_{b-e} \quad (1)$$

where contact resistance R_c between the heater and the bottom of the base plate surface is defined as

$$R_c = \frac{T_h - T_{bd}}{Q} \quad (2)$$

where T_h is the center temperature of the heater and T_{bd} is the center temperature at the bottom surface of the base plate.

Interface resistance $R_{b\text{-}e}$ between the evaporation section of the heat pipe and the top of the base plate surface is defined as

$$R_{b-e} = \frac{T_{bu} - T_e}{Q} \quad (3)$$

where T_{bu} is the center temperature at the top surface of the base plate and T_e is the center temperature of the evaporation of the heat pipe. Spreading resistance R_{sp} and conduction resistance R_m with unchanged material are constant. Therefore, $R_{b\text{-}e}$ is determined by R_e in the case where R_c, R_{sp}, and R_m are determined in advance.

As suggested by Faghri [2], Peterson [5], and Dunn and Reay [6], the thermal performance of a heat pipe can mainly be described by its effective thermal resistance and maximum heat transport capacity. The thermal resistance can be calculated by dividing the temperature difference between hot and cold ends by the corresponding heat throughput. In this study, R_{hp} is the thermal resistance of the heat pipe, which is calculated as

$$R_{hp} = \frac{T_e - T_c}{Q} \quad (4)$$

where T_e and T_c are the averaged temperature of the evaporation section and the condensation section, respectively. The Q is the corresponding heating power minus the heat loss, which is estimated at 5% in the experiments. According to the uncertainty analysis proposed by ISO standards [24], the uncertainties of temperature and thermal resistance measurement are ±0.5 °C and ±5%, respectively, and the uncertainties of heat inputs are within ±5% and ±2% between 0–10 W and 10–100 W, respectively.