A Study on the Heat Transfer Characteristics of a Self-Oscillating Heat Pipe

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In this paper, the heat transfer characteristics of a self-oscillating heat pipe are experimentally investigated for the effect of various working fluid fill charge ratios and heat loads. The characteristics of temperature oscillations of the working fluid are also analysed based on chaotic dynamics. The heat pipe is composed of a heating section, a cooling section and an adiabatic section, and has a 0. 002m internal diameter, a 0. 34m length in each turn and consists of 19 turns. The heating and the cooling portion of each turn has a length of 70mm. A series of experiments was carried out to measure the temperature distributions and the pressure variations of the heat pipe. Furthermore, heat transfer performance, effective thermal conductivity, boiling heat transfer and condensation heat transfer coefficients are calculated for various operating conditions. Experimental results show the efficacy of this type of heat pipe.

Key Words: Heat Transfer Characteristics, Self-Oscillating Heat Pipe, Fill Charge Ratio, Working Fluid, Effective Thermal Conductivity, Heat Transfer Performance, Boiling Heat Transfer Coefficient, Power Spectrum

Nomenclature

A : Flux area of the heat pipe[m^2]
A_B : Heat transfer area of a heating section[m^2]
A_C : Heat transfer area of a cooling section[m^2]
g : Acceleration due to gravity[m/s^2]
h_B : Mean boiling heat transfer coefficient [W/m^{2}K]
h_C : Mean condensation heat transfer coefficient [W/m^{2}K]
k_{eff} : Effective thermal conductivity [W/mK]
L : Length of heat pipe [m]
N : Raw number of heat pipe
P : Pressure [Pa]
Q : Heat load [W]
q : Heat flux [W/m^2]
r_{MAX} : Max. diameter of pipe [m]

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T_C : Mean temperature of a cooling section[℃]
T_E : Mean temperature of a adiabatic section [℃]
T_H : Mean temperature of a heating section[℃]
α : Fill charge ratio of working fluid [%]
ρ_l : Liquid density [kg/m^3]
ρ_v : Vapor density [kg/m^3]
σ : Surface tension [N/m]

1. Introduction

The increasing technological demands of today call for complex systems, which in turn require sophisticated electrical and electronic devices. The continuing trend toward miniaturization and generation of greater power is well known and the amount of heat generated in these electronic devices is increasing. However, it is very difficult to arrange a cooling device properly near the heating element so that it can rapidly get rid of the heat generated.

It is an important matter to effectively transfer heat to a cooling section from micro-electronic devices (Kim et al., 1998a, 1998b). Also, there is a need for a simple, reliable, highly efficient heat transport device to resolve heat problems arising in outer space, such as the radiation of heat in space equipment, satellites, and robots.

There are a number of different heat transport devices in use for eliminating heat. The most common device is a heat pipe that returns condensate fluid by capillarity. The heat pipe has a wick composed of a porous substance; it is maintained at a vacuum and is charged with a working fluid (Peterson, 1994 ; Faghri, 1995 ; Boo, 1998).

Even though the heat pipe was developed only recently, it has been adopted by a variety of users. But existing heat pipes have some drawbacks. They need a wick with complex features in order to reflux condensate fluid, and their performance deteriorates due to the blending of anti-condensate gases as air. Furthermore, with such a design it is difficult to produce a heat pipe with a fine diameter.

The self-oscillating heat pipe consists of three sections (Akachi, 1988): a heating section, an adiabatic section, and a cooling section. The inside is charged with a fixed amount of working fluid. After the fluid is evaporated and expanded in the heating section of the heat pipe, bubbles of steam become smaller or are destroyed while they pass between the adiabatic and the cooling sections; because of this, pressure waves are generated in the working fluid (Akachi, 1990 ; Koizumi, 1992). As the working fluid and bubbles circulate actively inside the heat pipe and produce axial direction vibration, heat transport is accelerated. The self-oscillating heat pipe has no wick at all, which distinguishes it from other heat pipes. Therefore, it is desirable that bubbles in the heat pipe should not be destroyed, but rather should become "plug flow", as they take the form of a piston and gain enough driving force after nucleate boiling in the heating section of the heat pipe.

Hosoda et al. (1997) derived a formula for calculating the largest radius of a heat pipe which can generate a vapor plug as follows:

$$r_{MAX}=0.92\sqrt{\frac{\sigma}{g(\rho_l-\rho_v)}} \tag{1}$$

According to the above formula, based on saturated water at 313K, the greatest radius of the heat pipe is 2.7×10^{-3}m.

The self-oscillating heat pipe has a simple structure because it does not have a wick. Therefore, its production costs are reasonable, and it can be smaller in size than existing heat pipes. Due to these characteristics, we expect it will be useful for many practical applications.

Since its possibility was expressed around 10 years ago, some research has been performed on this type of heat pipe. Chandratilleke et al. (1996) used liquid nitrogen as the working fluid in a very low temperature environment with a heat pipe of inner diameter 2.4mm. Miyazaki et al. (1998) studied the pressure distribution inside a copper pipe of inner diameter 2.0mm by using R134a and R142b. Nishio et al. (1997) used copper pipes and glass pipes, both of inner diameter 2.5mm, and Maezawa et al. (1997) used copper pipes of inner diameter 1mm~2mm to investigate effective thermal conductivity and fluid flow characteristics of heat pipes. Through these basic investigations, it has been proven that the self-oscillating heat pipe has outstanding performance in heat transport.

The objective of this research is to obtain basic information about the self-oscillating heat pipe which can overcome some of the drawback of existing heat pipes. For this, a simple type of self-oscillating heat pipe is introduced, and its heat transfer characteristics are investigated for various fill charge ratios of the working fluid and for different heat loads.

2. Experimental Apparatus & Method

2.1 Experimental apparatus

A schematic diagram of the experimental apparatus is shown in Fig. 1. In this figure, ⑥ is the self-oscillating heat pipe, ⑦ is a pen recorder, ⑧ is a PC, ⑪ are C-A thermocouples whose wire has a diameter of 50 μm, ② is a pressure transducer

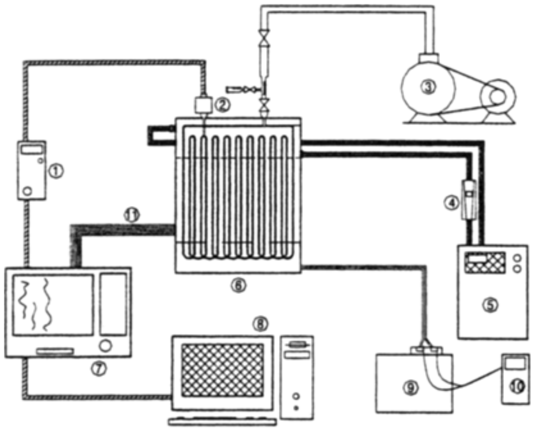

Fig. 1 Schematic diagram of experimental apparatus

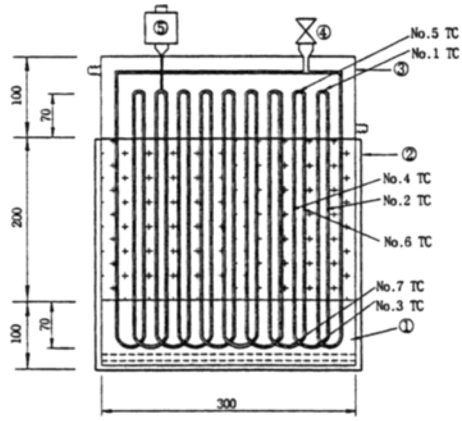

Fig. 2 Drawing of the self-oscillating heat pipe

for measuring internal pressure of the heat pipe, and ⑨ is a voltage regulator that controls heat load for the heating section of the pipe. Also, ⑩ is a multimeter for calculating heat load, ⑤ is a water bath for maintaining cooling water temperature of the cooling section of the heat pipe, ④ is a flow meter to measure flow rate of cooling water, and ③ is a vacuum pump for maintaining a high vacuum inside the heat pipe.

Figure 2 is a detailed drawing of the self-oscillating heat pipe. The heat pipe uses distilled water as a working fluid. The heat pipe is made of copper pipe having an external diameter 0.0032m and internal diameter of 0.0020m. ①, ② and ③ are the heating section, the adiabatic section, and the cooling section, respectively. ④ is the working fluid filling port, and ⑤ is a pressure transducer for measuring the heat pipe's inner pressure.

The heating section is the part heated by an electrical heater which is located inside the copper block. The heater controls the heat load by a voltage regulator. The adiabatic section is insulated with glass wool to prevent heat loss, and the cooling section is made of an acrylic block. The cooling section is arranged so as to be cooled by cooling water from a water bath. In order to install the working fluid filling port and the pressure transducer, each linking part was manufactured. The height of the heat pipe is 0.34m, of the heating section 0.07m, of the adiabatic section 0.2m, and of the cooling section 0.07m.

2.2 Experimental method

In the experiment, the fill charge ratio of the working fluid, as a percentage of the inside volume of the heat pipe, was changed from 20 to 80% at 10% intervals. The heat load of the heating section was increased from 100W to 600W at 100W intervals. As shown in Fig. 2, the temperature of each part of the heat pipe was measured with C-A thermocouples: 2 for the heating section, 3 for the adiabatic section, and 2 for the cooling section. The entry temperature of the cooling water was adjusted to maintain 30℃ using a controlled water bath, and cooling water was supplied to the heat pipe at a rate of $2 \sim 4l/$min. Throughout the experiment, the largest temperature difference between entering and exiting water was about 2℃. A fill charge device was made and used in order to smoothly inject the working fluid into the heat pipe, and the inside of the heat pipe was maintained at a 0.5Torr vacuum using the vacuum pump. The measured amount of working fluid was injected into the heat pipe using the fill charge device. Also, pressure of the heat pipe was measured using a pressure transducer. Data were sampled at the sampling frequency of 3, 9 and 135Hz.

In order to increase reliability of experimental results, two experiments were performed under the same condition and the mean value of the resulting data was calculated. The standard deviation of the temperatures of the heating section was 2.0℃ and that of the cooling section was 0.6℃. When changing from one fill charge ratio to another, the working fluid was injected when the heat pipe was completely cooled. Data were measured after the heat pipe had reached a steady state. Because anti-condensate gas existing inside the heat pipe can affect heat transport characteristics, air was thoroughly eliminated before charging with the working fluid. To charge the heat pipe with the correct amount of working fluid, we attempted to achieve reliability through several replications. Since the inside volume of heat pipe is $2.234\times10^{-5}m^3$, the volume of working fluid used for charging was estimated based on this volume.

3. Experimental Results and Discussion

3.1 The effect of fill charge ratio and heat load on temperature and pressure variation

Figure 3 shows the temperature changes for 100 seconds in the heating and cooling sections at a fill charge ratio of 30% and a heat load of 400W. It can be seen that the temperature changes in an irregular fashion. Such fluctuations are mainly due to bubble behavior, circulation and vibration of the working fluid causing pressure wave motion.

Figures 4 and 5 show the effect of fill charge ratio on mean temperature of the heating and the cooling sections, respectively. Mean temperature of the cooling section is not very sensitive to fill charge ratio. On the other hand, mean temperature of the heating section increases very rapidly as the heat load goes up, especially at higher fill charge ratios. It shows that the heat pipe approaches its working limit easily as heat load goes up at higher fill charge ratios, and it means that heat transport ability declines rapidly when the fill charge ratio of the working fluid exceeds a certain value.

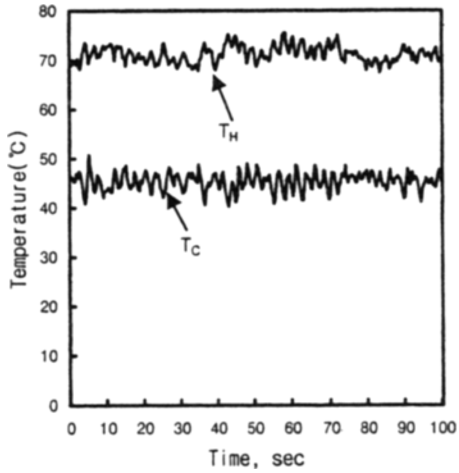

Fig. 3 Temperatures of the heating and the cooling sections

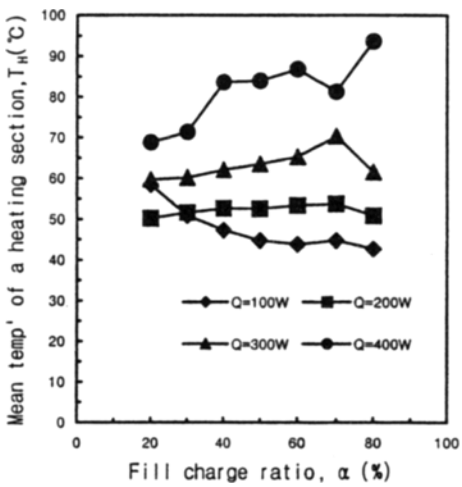

Fig. 4 Mean temperatures of the heating section versus fill charge ratio

Figure 6 shows the mean inner pressure of the heat pipe for various heat loads, based on fill charge ratio. As shown in the figure, inner pressure is less sensitive to fill charge ratio at lower heat loads. However, when the heat load exceeds a certain value, it can be seen that the

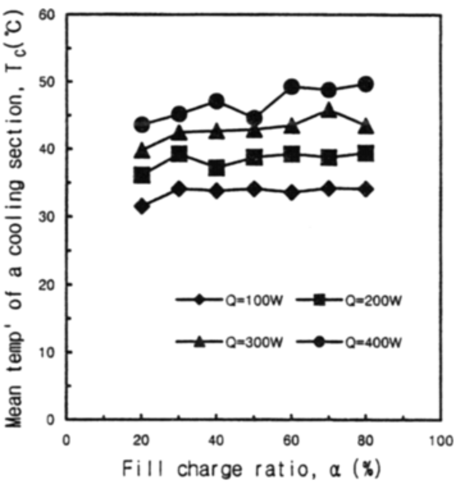

Fig. 5 Mean temperatures of the cooling section versus fill charge ratio

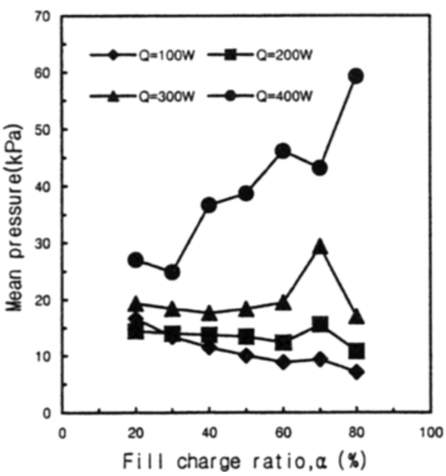

Fig. 6 Mean inner pressures of the heat pipe versus fill charge ratio

degree of vacuum in the heat pipe decreases rapidly as the fill charge ratio increases. This explains the fact that when the heat load is continually increased, the heat pipe reaches its working limit first at higher fill charge ratios. Actually, when the heat load was 600W, the vacuum of the heat pipe broke at fill charge ratio over 60%, and the heating section temperature increased rapidly so that it was impossible to continue the experiment. Similarly, when the fill charge ratio is increased, the inner pressure of the heat pipe rapidly increases at a given heat load until it

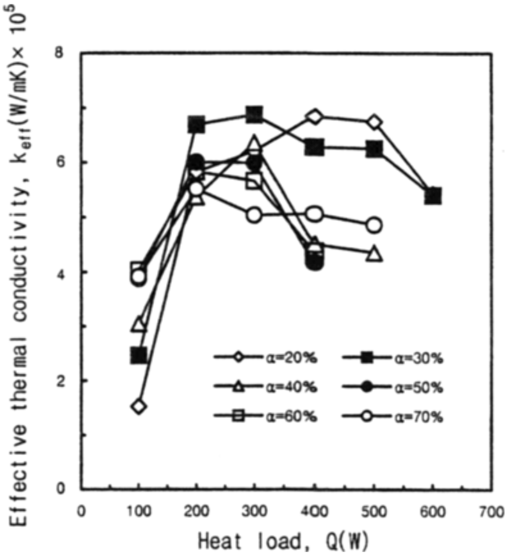

Fig. 7 Effective thermal conductivity of the heat pipe versus heat load

reaches its working limit. Specifically, as the fill charge ratio exceeds 60%, lack of inner space in the heat pipe and flow inefficiency of the working fluid contribute to malfunction even at low heat loads. In this work, as shown in Fig. 6, the absolute inner pressure range of the heat pipe is about 10~60kPa under normal conditions.

3.2 Heat transfer characteristics of the self-oscillating heat pipe

Figure 7 shows the variation of effective thermal conductivity k_{eff} as a function of heat load Q. Effective thermal conductivity was calculated as follows. To get mean temperature T_H of the heating section and T_C of the cooling section, temperatures were measured and averaged. For this, No.3 and No.7 thermocouples for the heating section and No.1 and No.5 thermocouples for the cooling section were used as depicted in Fig. 2.

$$Q=NAq=NAk_{eff}\frac{(T_H-T_C)}{L} \tag{2}$$

where Q is total heat load, q is heat flux, and L is the length of the heat pipe, that is, the distance between T_C measuring point and T_H measuring point. N is the number of lines of the heat pipe (20 in this case). A denotes the flux area of the inner part of the heat pipe.

In Fig. 7, effective thermal conductivity k_{eff} does not show much change as fill charge ratio is varied. However, when the fill charge ratio is below 50%, the effective thermal conductivity k_{eff} becomes high in general. On the other hand, when the fill charge ratio is more than 60%, the effective thermal conductivity shows a tendency to decrease rapidly under the same operating conditions. These results are analogous to those obtained in Figs. 4, 5, and 6. That is, when the fill charge ratio is higher than 60%, the temperature difference between the heating and the cooling sections increases rapidly as the heat load exceeds 400W. Also, the degree of vacuum of the heat pipe drops very rapidly. Thus, if the fill charge ratio of the working fluid exceeds a certain amount, the lack of inner space in the heat pipe and inefficiency in the flow of the working fluid cause the flux speed of the working fluid and the bubble that circulate inside the heat pipe to slow down. Therefore, heat transfer efficiency suddenly drops.

The effective thermal conductivity of the heat pipe in Fig. 7 is 1000~2000 times greater than the thermal conductivity of copper. Thus, this type of heat pipe has excellent heat transport characteristics. However, there has been little research so far on the effective thermal conductivity of the self-oscillating heat pipe. Results obtained by Hosoda et al. (1997) demonstrated that the range of the effective thermal conductivity was $10^5 \sim 10^8$ W/mK with a copper pipe of inner diameter 2.0mm. Maezawa et al. (1997) also found a range of effective thermal conductivity of $2 \times 10^4 \sim 10^5$ W/mK. In their study, copper pipes with inner diameter of 1.0, 1.6 and 2.0mm were used, and the condensation section was cooled by air.

Figure 8 shows mean boiling heat transfer coefficient h_B as a function of heat flux q. When the heat pipe works ideally, heat transfer is very vigorous. Accordingly, both the evaporation and the condensation sections can be treated as isothermal. Besides, since the copper pipe used has a large thermal conductivity and a thickness of only 0.0006m, the temperatures of the inside and the outside of the pipe are almost the same. In fact, according to numerical analysis using

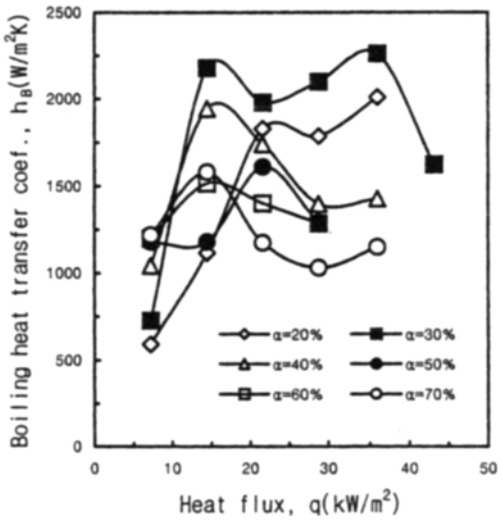

Fig. 8 Boiling heat transfer coefficients of the heat pipe versus heat flux

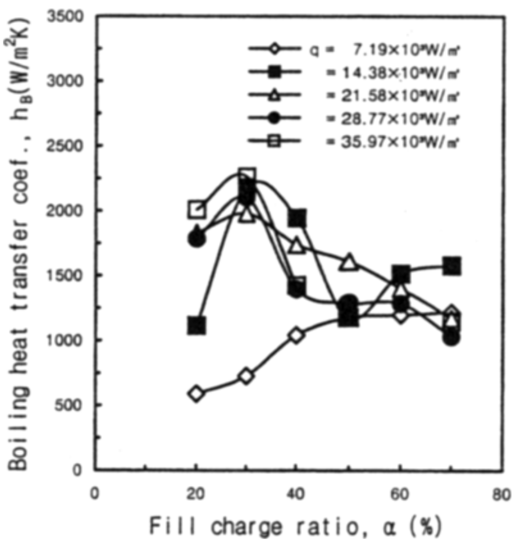

Fig. 9 Effects of fill charge ratio on boiling heat transfer coefficients

Patankar's TDMA method (Patankar, 1980), the difference between the inside and the outside temperature of the heat pipe was under 0.5℃. Therefore, the temperatures of the inner surface of the heat pipe can be considered the same as the temperature of the outer surface as measured with C-A thermocouples. Accordingly, we used T_H, T_C and T_E to calculate the mean boiling heat transfer coefficient h_B and the mean condensation

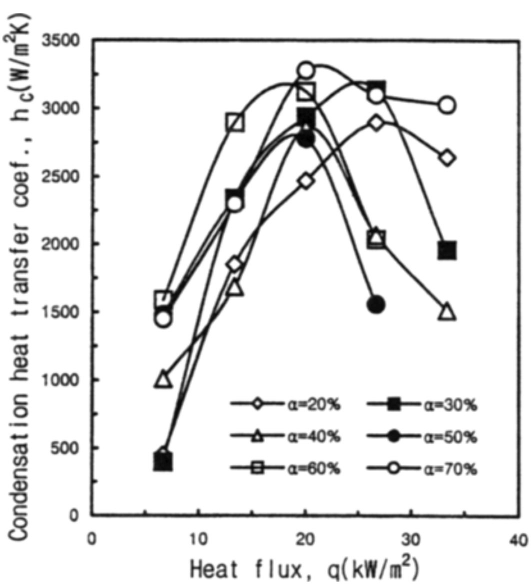

Fig. 10 Condensation heat transfer coefficients of the heat pipe versus heat flux

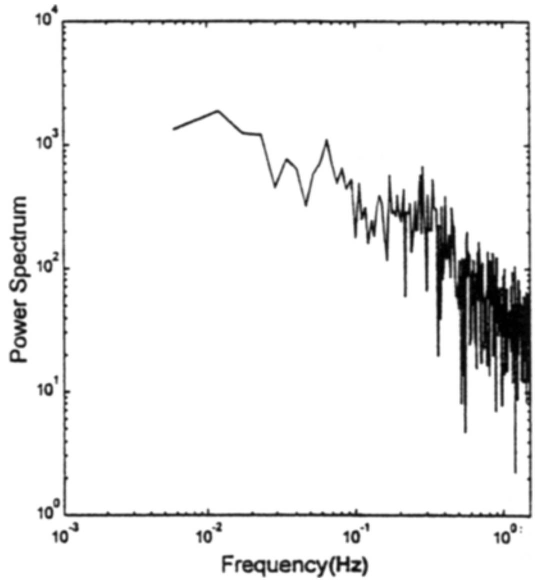

Fig. 11 Power Spectrum by Fourier Transform(α= 30%, Q=400W)

heat transfer coefficient h_C as in Eq. (3) and (4) respectively. The mean temperature of the adiabatic section T_E is the mean value from No. 2, No. 4, and No. 6 of Fig. 2. A_B and A_C are the heat transfer areas of the heating and the cooling sections.

$$h_B = \frac{Q}{A_B(T_H - T_E)} \tag{3}$$

$$h_C = \frac{Q}{A_C(T_E - T_C)} \tag{4}$$

Figure 9 shows mean boiling heat transfer coefficient as a function of fill charge ratio. In general, the mean boiling heat transfer coefficient appears high when the fill charge ratio varies from 20 to 40% except when heat load is too low. This is the same trend as shown in Fig. 8, and it means that when using the self-oscillating heat pipe, it is desirable to maintain a working fluid fill charge ratio of 30%.

Figure 10 shows how the mean condensation heat transfer coefficient h_C varies with heat flux q. At first glance, the fill charge ratio appears to have little effect. However, the heat pipe reaches its working limit most rapidly when the fill charge ratio is 70%. This is because at high fill charge ratio the inner space in the heat pipe is too small for the working fluid vapor to circulate actively between the heating and the cooling sections. Therefore, the vacuum in the heat pipe is destroyed easily and the pipe's heat transport ability drops rapidly. In this case, the largest heat flux was about 40kW/m^2 just before reaching the working limit at a fill charge ratio of 70%. But, at smaller fill charge ratios, the pipe worked normally over a wider range of heat fluxes. Consequently, the condensation heat transfer coefficient is not greatly affected by the fill charge ratio. Also, as the fill charge ratio increases greatly, the inner pressure increases rapidly because of the lack of inner space, and the heat pipe reaches its working limit even at small heat fluxes. Therefore, with this type of heat pipe, it is not desirable to use fill charge ratios higher than 40%.

3.3 Internal behavior characteristics of working fluid in the self-oscillating heat pipe

The power spectrum provides important information regarding the random characteristics of a system. Figrue 11 shows the power spectrum of temperature oscillations when the fill charge ratio is 30% and the heat load is 400W, and has a

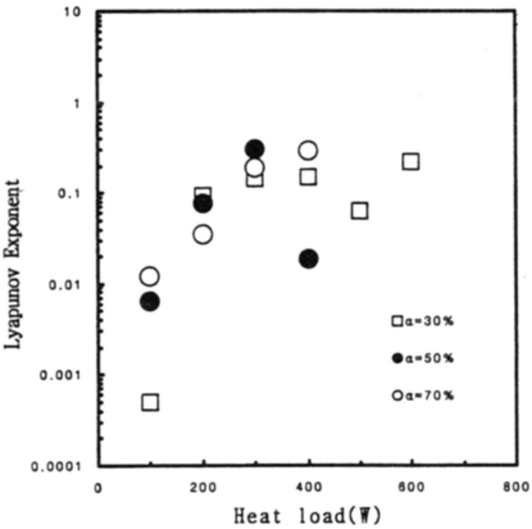

Fig. 12 Lyapunov exponent

1/f type spectrum. It means that the oscillation assumes chaotic complexity.

Figure 12 shows how the Lyapunov exponent varied with heat loads. The Lyapunov exponent must have a positive value for the inner pipe to be considered random. As shown in Fig. 12, its value is positive for all fill charge ratios. Also, the Lyapunov exponent in the figure goes up as heat load increases. However, the fill charge ratio does not have much effect on the Lyapunov exponent. This leads to an increase of instability inside the heat pipe in accordance with an increase of heat transport. The result of Fig. 12 may be due to the existence of chaotic behavior in bubble generation, growth and collapse in the heat pipe.

4. Conclusion

From a series of experiments, the following results were observed.

(1) The range of effective thermal conductivities is $5\times10^5\sim7\times10^5$W/mK in the heat pipe, which is 1000~2000 times the thermal conductivity of copper.

(2) The boiling heat transfer coefficient of the heat pipe is the largest when the fill charge ratio is 30%. In this case, the range of boiling heat transfer coefficients is about 2000~2300W/m^{2}K.

(3) The fill charge ratio does not have much effect on the condensation heat transfer coefficient. As the fill charge ratio increases beyond 60%, lack of inner space in the heat pipe and flow inefficiency of the working fluid contribute to malfunction even at low heat fluxes.

(4) The inside of the heat pipe was influenced by chaotic dynamics, and as heat load increases, the Lyapunov exponent also increases. This means that the inside of the heat pipe is approaching a condition of complex chaos with increases of instability.

Reference

Akachi H., 1990, "Structure of a Heat Pipe," *United States Patent*, Patent No. 4921041.

Akachi H., 1988, "The Loop Type Capillary Heat Pipe," *Japanese Patent* (*A*), 1988-318493.

Boo J. H., 1998, "Fundamental Principle and Design Sequence of Heat Pipes," *Proceedings of the KSME* 1998 Annual Meeting, pp. 3~18.

Chandratilleke G. R., Ohtani Y., Hatakeyama H. and Nakagome H., 1996, "Development of Looped Heat Pipes for Cryogenic Application," *JSME Symposium* (VI), pp. 140~143.

Faghri A., 1995, "Heat Pipe Science and Technology," Taylor & Francis.

Hosoda M., Nishio S. and Shirakashi R., 1997, "Study of Meandering Closed-Loop Heat Transport Device," *Transaction of the JSME* (*B*), Vo. 64, No. 622, pp. 1845~1851.

Kim W. T., Song K. S. and Lee Y., 1998a, "Design of a Two-Phase Loop Thermosyphon for Telecommunications System (I)," *KSME International Journal*, Vol. 12, No. 5, pp. 928~941.

Kim W. T., Song K. S. and Lee Y., 1998b, "Design of a Two-Phase Loop Thermosyphon for Telecommunications System (II)," *KSME International Journal*, Vol. 12, No. 5, pp. 942~955.

Koizumi, 1992, "A Cooler and a Thermal Control Device," *Japanese Patent* (*A*), 1992-20788.

Maezawa S., Izumi T., Nakajiwa R. and Gi K., 1997, "Nonliner Chaotic Characteristics of Oscillating Heat Pipe," *34th Heat Transfer*

Symposium of Japan, pp. 275~276.

Miyazaki Y., Akachi H. and Arikawa M., 1998, "Study on Oscillating Heat Pipe," *35th Heat Transfer Symposium of Japan*, pp. 531~532.

Nishio S., Hosoda M., Nagata S. and Watanabe K., 1997, "Bubble-Driven Heat Transport Tube," *34th Heat Transfer Symposium of Japan*, pp. 269~270.

Patankar V. S., 1980, "Numerical Heat Transfer and Fluid Flow," *McGraw-Hill*.

Peterson G. P., 1994, "An Introduction to Heat Pipe," *John Wiley & Sons*, Inc.