On the operating temperature of heat pipes

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Abstract. The relationship between the *operating temperature* of heat pipes and the maximum heat transport capacity posed by the heat pipe capillary limit is often overlooked. It is demonstrated through heat pipe experiments that for a given heat input, there exists a minimum temperature for the heat pipe system to operate. This phenomenon occurs due to the temperature dependence of the thermo-physical properties of the working fluid in the heat pipes and the working temperature range of the heat pipe system can thus be discerned by the capillary limit equation in conjunction with the heat pipe transient equation obtained by energy conservation. It may sometimes seem counterintuitive in the sense that if a heat pipe system is aided by a fan (and therefore increase the heat transfer coefficient), then the heat pipes break down and reduce the effectiveness of the thermal management system. This is due to the fact that heat pipes have excessively high *effective* thermal conductivity and their breakdown leads to heat transfer only through their constituent materials, whose thermal conductivities are lesser by at least an order of magnitude. Heat pipes in a thermal management system must therefore be meticulously designed for precise temperature ranges.

1. Introduction

Heat pipes have found their applications in circumstances where dissipating large amounts of heat from relatively small areas is necessary. They have excessively high *effective* thermal conductivity and exhibit very low temperature differences along its length (nearly isothermal profile) resulting in high thermal efficiencies. Current scientific literature on the principle of heat pipes is nearly exhaustive with numerous text books [1, 2, 3] written; and extensive attention has also been given for its technological development. However, a fundamental issue regarding the operating temperature of heat pipes is either unaddressed or overlooked, which we aim to address.

Ideally, as the heat pipe must exhibit isothermal temperature profiles along its length, the theoretical operating temperature of the heat pipe can be determined by lumped parameter analysis, in addition to using the material properties (specific heat of heat pipe material, surface area exposed to heat transfer, specific heat of wick material and specific heat of the working fluid) and environmental properties (heat transfer coefficient and ambient temperature). This operating temperature is thus a parameter of design and environmental constraints.

The maximum heat transport capacity of a heat pipe is limited by its working fluid and its internal wick structure. This limit is considered on the basis of the thermo-physical properties of the working fluid which creates the required capillary pressure for the working fluid to return back to the evaporator. Any heat supplied beyond the capillary limit would lead to the breakdown of the heat pipe resulting in a non-isothermal temperature profile. As the thermo-physical properties of the working fluid are temperature dependent, the maximum heat transport capacity would correlate to a specific working fluid temperature. This working temperature can therefore be regarded as a limit posed by the thermo-physical properties of the working fluid.

Consequently, if the working temperature of the heat pipe obtained from lumped capacitance analysis is below the minimum working temperature defined by the thermo-physical properties of the working fluid, then the heat pipe is said to have broken down. We demonstrate this in a heat pipe experiment by controlling its *operating temperature* (refer section 2.1) using a fan (and thereby change the external heat transfer coefficient).

Needless to say, we criticise all studies that only address the capillary limit aspect of heat pipes and disregard its operating temperature. For instance, Brusly Solomon et al. [4] have proposed a method to determine the *effective* thermal conductivity of heat pipes using the capillary limit equation. However, neither have they computed the maximum heat transport capacity of their heat pipe nor have they justified the occurrence of temperature differences of over 50°C along the length of their heat pipe, which clearly indicates a breakdown.

2. Governing equations

The theory of heat pipes is well established and we make an attempt only to briefly state the fundamentals. We first derive an expression for heat pipe transient (temperature as a function fo time) by applying energy conservation and later examine the capillary limit of the heat pipe, whose relation to the heat pipe transient is the centre of attention in the present work.

2.1. Heat pipe transients - lumped parameter analysis

The transient response of a heat pipe is analysed using a lumped analytical model derived from basic energy balance for a control volume using general capacitance analysis [5]. In the present study, the radiative heat transfer effect is assumed to be negligible. The general lumped analysis with convective heat transfer from the surface is given by

$$Q_{in} - hA(T - T_\infty) = C_t \frac{dT}{dt}, \tag{1}$$

where Q_{in} is the heat input, h is the convective heat transfer coefficient, A is the area exposed for convection, C_t is the total thermal capacity of the heat pipe system, T is the heat pipe system temperature, T_∞ is the ambient temperature and t is time. It is worthy to note that the total thermal capacity C_t is to be calculated as a volume fraction weighted average of the thermal capacities of the constituents of the heat pipe system including the barrel, wick structure and the working fluid.

If we define the convective thermal resistance to be

$$R_c = \frac{1}{hA}, \tag{2}$$

a transient solution for eq. (1) could be obtained with appropriate boundary conditions for the start-up as well as the shut-down of the heat pipe system. The solutions for eq. (1) would have the form

$$T(t) = T_\infty + Q_{in} R_c \bigg(1 - \exp\Big(\frac{-t}{C_t R_c}\Big)\bigg) \tag{3}$$

for the start-up and

$$T(t) = T_\infty + (T_0 - T_\infty)\exp\Big(\frac{-t}{C_t R_c}\Big) \tag{4}$$

for the shut-down from an initial temperature T_0. Subsequently, it can be construed from eq. (3) that a steady-state temperature of nearly $T_\infty + Q_{in}R_c$ is attained by the heat pipe system during the start-up, which we shall term as the *operating temperature* of the heat pipe.

2.2. Capillary limit

The capillary heat transport limit of a heat pipe is limited by the characteristics of the wick used and its interaction with the working fluid. Once the capillary limit is reached, any further increase in heat input would result in the breakdown of a heat pipe. This phenomenon can be quantified by the capillary pressure balance equation which, when written in its standard notation [3], is

$$\Delta P_{cap} + \rho_l g L \geq \int_{L_{eff}} \frac{\partial P_v}{\partial x} dx + \int_{L_{eff}} \frac{\partial P_l}{\partial x} dx + \Delta P_{e_{phase}} + \Delta P_{c_{phase}} + \Delta P_\perp + \Delta P_\parallel. \tag{5}$$

Moreover, eq. (5) can be further reduced to

$$\frac{2\sigma}{r_c} + \rho_l g L \geq \frac{16\mu_v L_{eff}\dot{Q}}{2r_v^2 A_v \rho_v h_{fg}} + \frac{\mu_l L_{eff}\dot{Q}}{\kappa A_l \rho_l h_{fg}} + \rho_l g d_v \tag{6}$$

with the following assumptions [3]: that the pressure gradient due to phase change in evaporator and condenser sections is negligible; and that the viscous losses are accounted for but the inertial effects are neglected in the vapour phase. The effective length L_{eff} in eq. (6) is a parametrised effective heat pipe length derived from them individual lengths of the evaporator, condenser and adiabatic sections of the heat pipe as

$$L_{eff} = \frac{L_e + L_c}{2} + L_a. \tag{7}$$

The maximum heat transport capacity limited by the capillary pressure can now be obtained from eq. (6) by plugging in the values of the thermo-physical properties (refer table A1 and table A2) of the working fluid and other heat pipe design variables. This heat transport limit, as mentioned earlier in section 1, is sensitive to changes in the *operating temperature* and increases with the increase in temperature of the working fluid. To paraphrase therefore, for a given heat input, there exists a minimum temperature only above which the heat pipe does not succumb to the capillary limit.

3. Experiment

3.1. Experimental setup

Our experimental setup consists of 16 gravity assisted copper heat pipes in parallel, heating element, thermocouples, data logger, power supply and the external experiment chamber. Each heat pipe is 85 mm long with a 6.5 mm outer diameter and lacks an adiabatic region. A stainless-steel mesh is used as a wick with 42.25% porosity and 3.1326e-11 m^2 permeability. The permeability of the wick is calculated using the modified Blake-Kozeny equation [6]. The working fluids used in this study are methanol (working range: 283K to 403K [6]) and ethanol (working range: 273K to 403K [6]) which are compatible with copper. The heat pipes are evacuated to a gauge vacuum pressure of 22 inHg which is verified using a compound pressure gauge. 14 ml of working fluid is charged into the heat pipe (35% fill ratio). To record the temperature data, a set of 14 T - type thermocouples ($\pm 0.5°C$), placed along the length of heat pipes (bottom, middle and top on random heat pipes) are used in conjunction with a data logger. All thermocouples are equally spaced along the length of the heat pipe and are mounted using Kapton tape. The data logger is set to log data at a sampling rate of 500 ms throughout the start-up, steady state and shut-down of the heat pipe. The heating element is an electric resistance heater with a maximum heat capacity of 200 W. To negate the effect of contact heat resistance between the surfaces of the heat pipes and the heater, a thin layer of thermally conductive heat sink compound (Anabond 652c™) is applied. Experiments are carried out in a stable lab temperature of 25°C.

3.2. Methodology

The maximum heat transport capacity, as discussed in section 2.2, is calculated for a range of the working fluid temperatures and tabulated (refer table 1). The heat pipes are then subjected to a controlled heat input (in W) using the resistance heater. It is ensured at first, that for the given heat input, the *operating temperature* of the heat pipe system exceeds the minimum temperature (discussed in section 2.2). In a following experiment, the heat pipe system is subjected to an increase in the convective heat transfer coefficient (by using a fan) for the same heat input.

Table 1. Heat transport limits calculated using capillary limit equation

Working fluid temperature (°C)	Maximum heat transport limit (W)	
	Methanol	Ethanol
50	45.12	20
70	52	26.15
90	55.68	32.97
110	59.79	38.37
130	66.03	43.94

4. Results and discussion

Following the methodology stated in section 3.2, experiments are conducted for heat inputs of 50 W and 60 W with methanol as the working fluid; and for heat inputs of 30 W and 40 W with ethanol as the working fluid. The results for these experiments, depicting the heat pipe transients with non-dimensional time on abscissa, are plotted in fig. 1, fig. 2, fig. 3 and fig. 4 respectively.

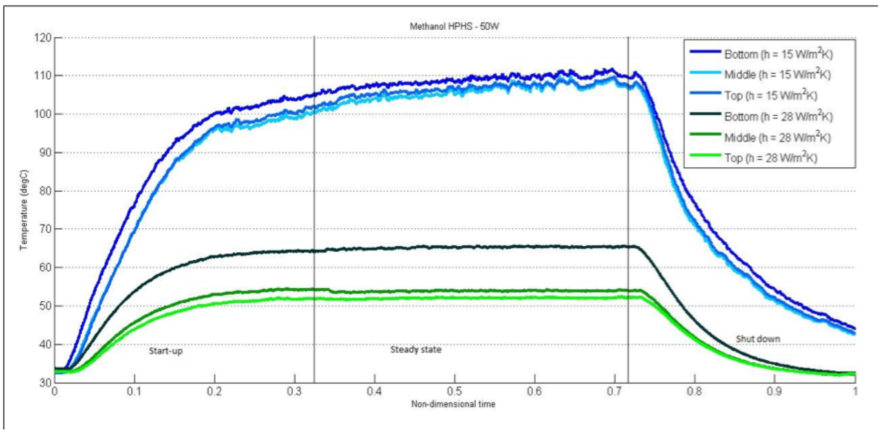

Figure 1. Comparison of temperature profiles of methanol heat pipe at various heat transfer coefficients for 50W heat input

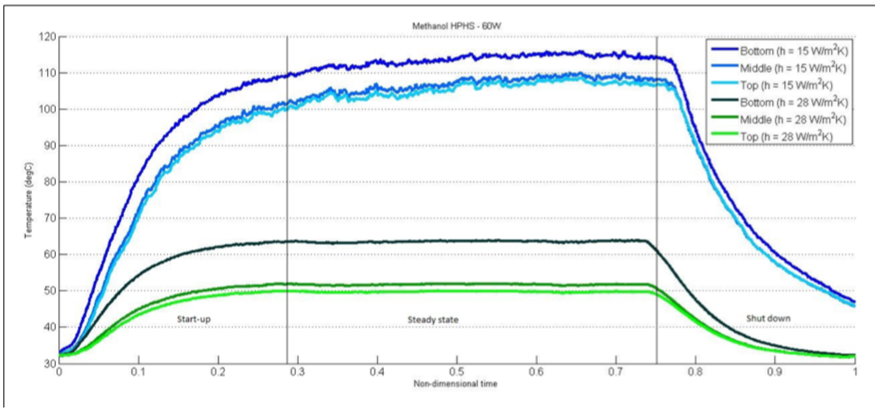

Figure 2. Comparison of temperature profiles of methanol heat pipe at various heat transfer coefficients for 60W heat input

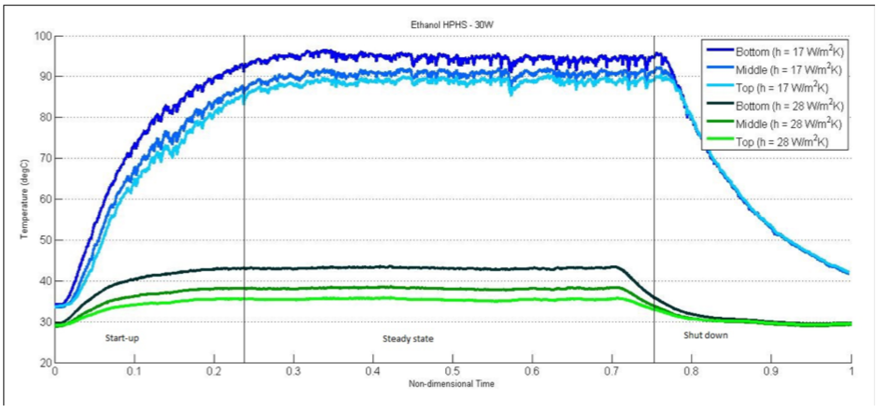

Figure 3. Comparison of temperature profiles of ethanol heat pipe at various heat transfer coefficients for 30W heat input

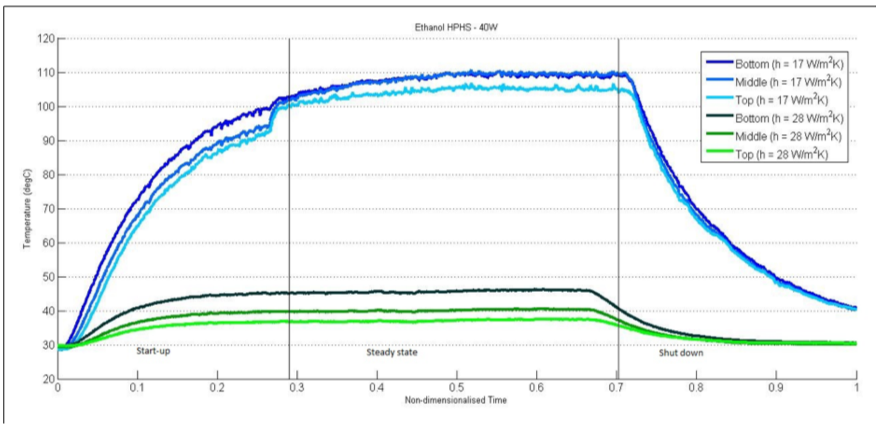

Figure 4. Comparison of temperature profiles of ethanol heat pipe at various heat transfer coefficients for 40W heat input

To demonstrate and strengthen our proposition, consider for instance, if a certain design requires the dissipation for 50W of heat, the minimum temperature required would be about 70°C (table 1). This is apparent through fig. 1, with 50W heat input, the maximum temperature achieved with a heat transfer coefficient of 15 $\mathrm{Wm^{-2}K^{-1}}$ (in the laboratory ambience) is 111°C and with the same heat input and a heat transfer coefficient of 28 $\mathrm{Wm^{-2}K^{-1}}$ the maximum operating temperature is 65°C. Referring to table 1, a minimum of 70°C operating temperature throughout is required to drive the heat pipe at 50W (capillary limit) which does not occur in the case where the heat transfer coefficient is higher (28 $\mathrm{Wm^{-2}K^{-1}}$). Clearly, as seen in fig. 1, the temperature distribution of methanol heat pipes, with a higher heat transfer coefficient, is not isothermal, indicating a break down.

5. Conclusion

This study elucidates the much overlooked relationship between the *operating temperature* of heat pipes and maximum heat transport capacity that is limited by the capillary structure (wick). For a given heat input, it is necessary that a minimum temperature is attained and an additional increase (deviation) in the heat transfer coefficient, from the initial design, rather inhibits the heat pipe action. This phenomenon is counterintuitive in the sense that if a heat pipe system is aided by a fan, then the heat pipes break down and reduce the effectiveness of the thermal management system. Heat pipes in a thermal management system must therefore be meticulously designed for precise temperature ranges as clarified in Marshburn's comprehensive technical report [7].

Author contributions

PS, VPR, VV and SVB identified the deficiency in literature regarding the *operating temperature* and conducted the experiments under the guidance of KNS. RNR designed and manufactured the heat pipe apparatus. All authors contributed equally in writing the manuscript.

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Appendix

Table A1. Thermo-physical properties of methanol [3]

Tempe -rature (°C)	P_v Saturation pressure (10^5Pa)	h_{fg} Latent heat (kJ/kg)	ρ_l Liquid density (kg/m^3)	ρ_v Vapour density (kg/m^3)	μ_l Liquid viscosity ($10^3\times$ Ns/m^2)	μ_v Vapour viscosity ($10^3\times$ Ns/m^2)	k_l Liquid thermal conduc-tivity (W/mK)	σ Surface tension ($10^2\times$ N/m)	C_{pv} Vapour specific heat (kJ/kgK)
50	0.55	1125	764.1	0.77	0.3990	0.0104	0.202	2.01	1.54
70	1.31	1085	746.2	1.47	0.314	0.0111	0.201	1.85	1.61
90	2.96	1035	722.4	3.01	0.259	0.0119	0.199	1.66	1.79
110	4.98	980	703.6	5.64	0.211	0.0126	0.197	1.46	1.92
130	7.86	920	685.2	9.81	0.166	0.0131	0.195	1.25	1.92

Table A2. Thermo-physical properties of ethanol [3]

Tempe -rature (°C)	P_v Saturation pressure (10^5Pa)	h_{fg} Latent heat (kJ/kg)	ρ_l Liquid density (kg/m^3)	ρ_v Vapour density (kg/m^3)	μ_l Liquid viscosity ($10^3\times$ Ns/m^2)	μ_v Vapour viscosity ($10^3\times$ Ns/m^2)	k_l Liquid thermal conduc-tivity (W/mK)	σ Surface tension ($10^2\times$ N/m)	C_{pv} Vapour specific heat (kJ/kgK)
50	0.29	872.3	762.2	0.72	0.72	0.0097	0.166	2.31	1.51
70	0.76	858.3	743.1	1.32	0.51	0.0102	0.165	2.17	1.58
90	1.43	832.1	725.3	2.56	0.37	0.0107	0.163	2.04	1.65
110	2.66	786.6	704.1	5.17	0.28	0.0113	0.160	1.89	1.72
130	4.3	734.4	678.7	9.25	0.21	0.0118	0.159	1.75	1.78