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Contribution to Heat and Mass Transfer for Space Experiments

THÈSE DE DOCTORAT EN SCIENCES DE L'INGÉNIEUR

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Chapter 1 Introduction

In our World, heat dissipation is a topic of utmost importance in most modern machines and constructions that we produce and use in everyday life: houses, trains, cars, airplanes, power plants, computer and spacecraft. All these can be considered “heat generators” as they dissipate in the environment part of the power needed or generated, in the form of thermal energy. When exceeded heat power is not correctly dissipated, a consequent increase of temperature will quickly appear leading to the device damage or destruction. For this reason, cooling systems or heat exchangers able to transfer thermal energy from one zone to another are required to ensure thermal control during operation, reducing global thermal conductivity and the rising of hot spots. The so designed thermal control machine is named Heat Pipe (HP) and its development and application is in constant evolution, adapting to the rhythm of the technological progress. In this Chapter we provide an overview of the HP concept, its operation conditions and limits. This introduction is a preamble to the research presented in this manuscript, devoted to the study of an evolved commercial HP using Self-Rewetting Fluid (SRF) that may provides interesting advantages.

1.1 Heat Pipes

Heat Pipe can be considered the simplest heat exchanger never invented. It essentially consists of a closed metal tube structure partially filled with fluid where heat is transferred through phase change by the working fluid at the ends of the pipe. A Heat Pipe device consists of 3 parts: the evaporator, the condenser and the adiabatic region. The evaporator region is the zone in thermal contact with the heat source (“heat in” in the Figure below), the condenser region is the zone in contact with the external environment or a radiator (always at lower temperature with respect to the evaporator, “heat out” in the Figure) and, finally, the adiabatic region is the remaining zone between the evaporator and the condenser. The adiabatic zone is not design to exchange heat with the environment and in non-ideal case its contribution on the dissipation is negligible with respect to the heat transferred at the condenser side.

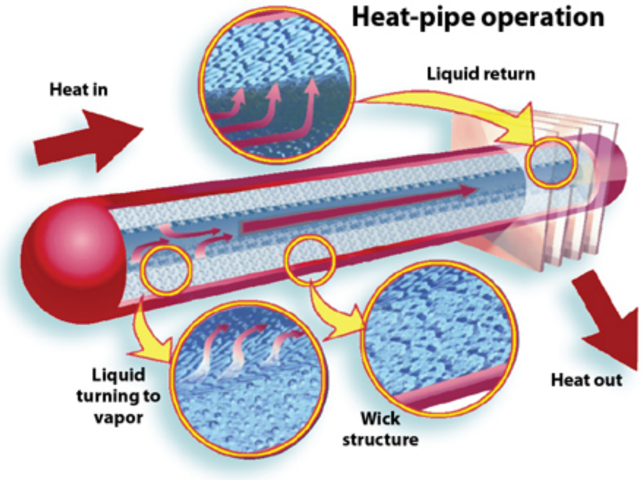

Figure 1.1: General concept of Heat Pipe operation [1].

Evaporation of the working fluid occurs at the evaporator side, the vapour flows through the pipe and releases thermal energy when it reaches the condenser region, kept at lower temperature. The liquid located at the condenser side is after pumped towards the evaporator by capillarity, thanks to a wick located at the internal side of the pipe structure (see Figure 1.1). The wick structure consists of a porous media that allows the return of the liquid from the condenser to the evaporator by capillary pressure. The process is passively activated when thermal gradient is set between the evaporator and the condenser; no external energy or other forces than the heat itself is required to pump out thermal energy from the heat source to the environment. In reality, HP activates when thermal gradient crosses a certain operation condition and if the gradient is too low, heat is transferred essentially by conduction through the pipe structure. The passivity of the HP is probably one of the most important advantage with respect to other available heat exchanger devices.

To correctly operate, HP is evacuated to at least medium vacuum quality level (normally between 1.33 to 10^{-1} Pa) before filling it with working fluid, to avoid the presence of non-condensable gases (air in general) into the heat transfer loop. If non-condensable gases are present during the operation of the HP, the gas is pushed by the vapour flow to the condenser side and it obstructs the working fluid vapour flow towards the condenser partially blocking the condensation process and so reducing the transferred heat to the external environment.

The use of HP technology is significantly more efficient than simply connecting condenser and evaporator regions with high conductive metal solid bar. In fact HP can reach thermal conductivity hundred times higher than an equivalent Copper rod of same section and length (401W/(m^{-2} K^{-1}) at 25°C) [2]. In general HP acts as a thermal superconductor with a very low thermal resistance R_T defined as:

$$R_T = \frac{T_e - T_c}{\dot{Q}} \tag{1.1}$$

where T_e and T_c are evaporator and condenser temperatures and $\dot{Q}$ is the heat power absorbed by the pipe. For commercial HPs in nominal operational regimes, standard values for thermal gradients between evaporator and the condenser are in the range of 1°C to 8°C [3]. Typical values (used for Copper/Water Heat Pipes) found in literature is 0.2 K W^{-1} cm^2 for the thermal resistances of the evaporator and condenser and 0.02 K W^{-1} cm^2 for axial thermal resistance, derived for unit area of the evaporator/condenser or pipe vapour section [3]. The temperature difference T_e - T_c will then be equal to

$$\Delta T = \dot{q_e} R_e + \dot{q_c} R_c + \dot{q_a} R_a \tag{1.2}$$

where $\dot{q_e}$, $\dot{q_c}$ and $\dot{q_a}$ are the heat fluxes through the evaporator, the condenser and the adiabatic region; R_e, R_c and R_a are the thermal resistances of the evaporator, condenser and adiabatic zone along the axis of the pipe. With this method, setting the input power to dissipate at the evaporator and the pipe dimensions, it is possible to obtain a good estimation of the maximum ΔT.

As an example, let's consider a 19 cm long Copper/Water Heat Pipe of regular geometry, with evaporator and condenser lengths of 4.5 cm, circular section of 4.4 cm diameter and 3.55 cm inner diameter for vapour space, dissipating a total power $\dot{Q}_{in}$ of 100 W. The evaporator heat flux $\dot{q_e}$ is obtained dividing the total power by the heat input area A_e ($\dot{q}_e = \dot{Q}_{in}/\mathrm{A}_e = 100\ \mathrm{W}/(4.5\ \mathrm{cm} \times 2\pi \times 2.2\ \mathrm{cm}) \approx 1.6\ \mathrm{W\ cm^{-2}}$) while the axial heat flux $\dot{q_a}$ is equal to the power divided by the cross sectional area occupied by the vapour $\mathrm{A}_{a,v}$ ($\dot{q}_a = \dot{Q}_{in}/\mathrm{A}_{a,v} = 100\ \mathrm{W}/(\pi\times(3.55\ \mathrm{cm})^2) \approx 2.5\ \mathrm{W\ cm^{-2}}$). Replacing $\dot{q_e}$, $\dot{q_c}$ and $\dot{q_a}$ and R_e, R_c and R_a (considering $\dot{q_e} = \dot{q_c}$ and $R_e = R_c$) into expression (1.2) a ΔT of only 0.69°C is obtained. Such a small ΔT is the consequence of the very high equivalent thermal conductivity of the HP device, thanks to the effect of heat transport phenomena by phase change. A reduced thermal gradient along the pipe also reduces the thermal stresses in the pipe structure itself minimizing the heat losses along the adiabatic zone. ΔT increases (slightly) increasing input power, as reported by

Table 1.

Type of conductor	Slope with power [K/W]
HP, D=4.4 cm	0.0085
HP, D=2 cm	0.025
HP, D=1 cm	0.07
HP, D=0.5 cm	0.22
Copper Rod, D=4.4 cm	0.31

Table 1: Slope comparison for HPs of different diameters with vapour area 78.41% of the pipe section area, pipe length of 19cm and at different input powers. In the table is reported also the case of a solid copper rod of same length.

The "positive force" (motive force) acting into HP is the capillary force pulling the liquid towards the evaporator. Capillary flow towards the HP evaporator is acting thanks to the presence of the wick structure at the internal side of the pipe structure. This force starts-up the pipe and guarantee the correct operation overcoming the dissipative and gravitational forces that oppose to the flow of the fluid into the pipe. In other words, the maximum capillary pressure in the wick, neglecting the pressure drops due to evaporation and condensation at the liquid-vapour interface [4], must be equal or greater than the total pressure drop in the pipe for the liquid to be able to flow in the porous media and to prevent evaporator dry out:

$$max(\Delta p_c) \geq \Delta p_l + \Delta p_v + \Delta p_g \tag{1.3}$$

In (1.3) Δp_c is the capillary pressure difference between evaporator and condenser regions, acting in the liquid present in the wick structure; Δp_l and Δp_v are the pressure viscous drops that oppose to the liquid and vapour motion and Δp_g is the gravitational contribution related to the inclination or morphology of the pipe. This last contribution is considered positive if it opposes the capillary force direction.

The wick structure is a generic terminology to indicate a porous or semi-porous media designed to promote capillarity of the fluid contained in it; depending on the HP structure and application, the wick structure can refer to a screen, sintered or grooved pipe as shown in Figure 1.2. The heat pipe is then named sintered heat pipe (SHP) and inner grooved heat pipe (IGHP) in these cases.

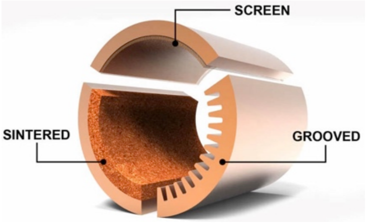

Figure 1.2: Different concept of possible wick structure on the internal side of the HP [5].

Capillary force manifests when the dimensions of the interstitials into which liquid flows are smaller than a critical length (referred to tube diameter or, in case of sintered and grooved structures it is sometimes referred to the equivalent diameter or liquid depth). Critical length value depends on fluid surface tension and different wick structures are used for different working fluids. The index used to compare different HP working fluids for comparable applications is named figure of merit and an more it is larger, more efficient is the binomial wick structure-working fluid on the overall operation of the considered heat pipe. This index is determined by physical characteristics of the fluid in the operational regime conditions (input power, temperature range, condenser temperature, etc.) and does not depend on the HP structure. Another parameter playing an important role on HP performances is the filling ratio; laboratory experiments showed that for different working fluids, when the percentage of the evaporator section volume occupied by the working fluids is increased to more than 85%, heat pipe performances decrease [6]. However, this value may change for Loop Heat Pipe (LHP) or Pulsating Heat Pipe (PHP) and in general, the optimal filling ratio is obtained experimentally once the wick structure and working fluid have been selected, and heat pipe nominal working conditions have been identified.

1.1.1 Applications

HPs are widely used in several high-tech and common applications: for instance in solar panel technology (Figure 1.3), floor heating system (Figure 1.4) or common circuit cooling system (Figure 1.5).

Figure 1.3: Solar panels can be considered the evaporator part of an HP.

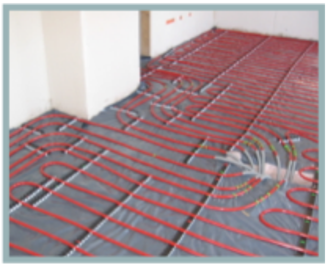

Figure 1.4: Floor heating system for domestic uses are designed as LHP.

Figure 1.5: Electronic cooling systems are normally Copper/Water HP.

Figure 1.6: Automotive subsystems require heat exchange devices to control temperature of mechanical parts and embedded electronics.

Figure 1.7: ISS is one of the best examples where HPs are massively used.

Figure 1.8: Rovers subsystems are all thermal controlled in order to work in hostile environments.

Figure 1.9: The development of new material makes possible to use HP as passive secondary emergency system for thermal control of power nuclear plants.

Figure 1.10: HPs are used to cool down underground soil layers to avoid permafrost melting during the warm season.

Figure 1.11: When strategic structure is built in not-optimal environmental condition, HPs are used to cool down the basement of the structure to avoid soil collapses.

With the increase of power requirements in informatics HP started to be used in all critical components in computer systems as CPUs (Central Processing Units) and GPUs (Graphic Processing Units). Later on, thanks to their low production cost and to the integration of mini-computer in all day life common devices, HP is also used for automotive subsystems (Figure 1.6). Of course, the use of HP in the aerospace field is massive and in particular, in space applications, where thermal dissipation is one of the most critical problematic for satellite, space stations (Figure 1.7) and rovers (Figure 1.8). The increase in performances for HP leads to use them also for less common applications as passive cooling system in power plants (Figure 1.9), permafrost cooling control application (Figure 1.10) and structure thermal stabilization (Figure 1.11). We certainly could say that HP development accompanied the last 40 years of technological progresses.

1.1.2 Limits

Heat Pipes are designed to optimally operate in specific conditions and indeed they operate less efficiently out from nominal condition. They even stop operating once they cross certain “limit conditions”. These limits are physically imposed by the fluid-thermal mechanism of the HP working fluid and wick structure geometry and define the operational area in the axial heat flux/ temperature graph where Heat Pipe can operate as shown in Figure 1.12.

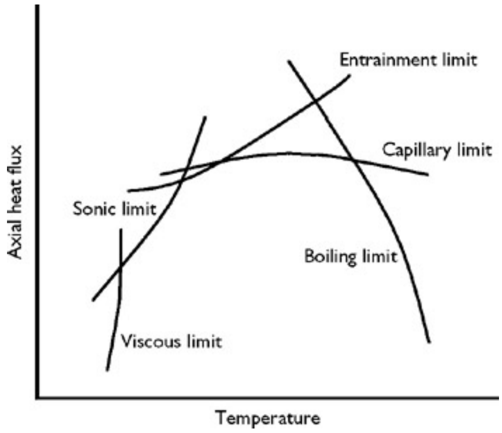

Figure 1.12: Heat Pipe limit conditions [2].

Referring to Figure 1.12, we notice five different limits that confine HP operational area; in general, we distinguish three main barriers that may contain one or more limits and they manifest when:

- temperature (and so pressure) is too low for the working fluid and not sufficient for vapour to flow up to the condenser side. The HP is in viscous regime and unable to activate (vapour pressure or viscous limit in Figure 1.12). This regime happens at low temperature condition, normally at pipe start-up when the pressure and temperature gradient between the evaporator and condenser is small. The maximum rate of heat transfer $\dot{Q}_v$ under this restricted operational regime is given by the following equation [7]:

  $$\dot{Q}_v = \frac{{D_v}^2 \lambda \rho_v p_v}{64 \eta_v L_{eff}} \tag{1.4}$$

  where D_v is the equivalent diameter of the vapour region, λ the latent heat of vaporization, ρ_v the vapour density, p_v the vapour pressure, η_v the vapour dynamic viscosity and L_{eff} the effective length (see Appendix A).

- Axial heat flux is too high and physical obstacles overcome the fluid circulation inside the HP (sonic limit, entrainment or flooding limit and capillary limit in Figure 1.12). Sonic limit appears when gas flowing to the condenser reaches sonic conditions: the generated shock wave increases temperature (and decreases pressure) acting as a cork in the vapour area with substantial interferences with the fluid circulation inside the HP porous media. The recommended maximum rate of heat transfer $\dot{Q}_s$ to avoid choked flow conditions (sonic limit) is given by [7]:

  $$\dot{Q}_s = 0.474\, A_v \lambda (\rho_v p_v)^{0.5} \tag{1.5}$$

  where A_v is the vapour passage area.

  Entrainment (or flooding) limit in Figure 1.12 is connected to interactions between liquid and vapour phase in the adiabatic region mainly. It consists of an instability appearing at the meniscus interface due to the flow condition of the vapour phase that is literally trapping small amount of liquids into the gas flow from the liquid interface. The dimensionless parameter that determines the onset of this phenomenon, measuring the relative importance of the fluid's inertia with respect to its surface tension is the Weber number *We*, defined as:

$$We = \frac{\rho_l v^2 L}{\gamma} \tag{1.6}$$

where v is the velocity of the flow and γ the surface tension. *We* is indicating when the vapour velocity is sufficiently high to produce shear forces on the liquid flow from the condenser to the evaporator generating waves at the liquid meniscus and causing possibly entrainment of liquid drops in the vapour flow. Entrainment will cause a starvation of fluid flow coming from the condenser and eventual "dry out" of the evaporator. The maximum heat transfer rate $\dot{Q}_E$ in this case is given by [7]:

$$\dot{Q}_E = A_v \lambda \sqrt{\frac{\rho_v \gamma}{L}} \tag{1.7}$$

where γ is the liquid surface tension and L the characteristic length of the wick structure.

The capillary or circulation limit in Figure 1.12 is instead expressed by equation (1.3) and will be largely argued in the next Chapter as basic concept to understand capillary force.

- Boiling limit is the last barrier that appears at high temperature regimes in Figure 1.12, when the pressure at the evaporator decreases below the boiling pressure of the fluid. Normally boiling limit is manifesting by local nucleation that appears when the evaporator HP structure temperature is higher than fluid boiling point; the fluid starts to boil forming eventually a stable vapour film between the liquid and the evaporator wall until dry out is reached. The presence of non-condensable gases in the HP working fluid area accelerates the boiling limit. The maximum heat flux under boiling condition is given by [7]:

$$\dot{q}_B = 0.12\lambda\sqrt{\rho_v}[\gamma g(\rho_l - \rho_v)]^{0.25} \tag{1.8}$$

Except for the *viscous limit*, heat pipes operating in regimes over the above-mentioned barriers are inexorably reaching dry out condition. “Dry out” corresponds to the partial or total drying of the evaporator region, caused by trapped liquid in the vapour zone or removed liquid from the wick structure (limits linked to high axial heat flux), or the evaporation rate is too high with respect to the liquid supplied (boiling limit for instance). When dry out occurs, temperature at the evaporator increases out of control until the HP metal structure itself reaches critical structural conditions, melting and gets destroyed and prior to this it is the thermal controlled apparatus to get destroyed of course. Dry out is a critical condition that must be avoided for any operational regime during any type of application.

1.2 Research on Heat Pipes

The interest in developing more efficient heat pipes is linked to numerous applications where compactness, reliability and efficiency are strictly required. The binomial HP material/working fluid is decisive for selecting right application types; for instance Titanium-alloy/Nitrogen Heat Pipes are used for cryogenic applications (< 30°K), Tungsten/Silver Heat Pipes operate at high temperature range (>2300°K) instead; in electronic cooling applications, where temperature must be controlled below 400°K, Copper/Water Heat Pipes are commonly used [3].

1.2.1 State of Art

Examples of studies aimed to increase HP performances consist in investigating several wick structure concepts as micro-channels [8] or researches on models selecting best working fluid candidate with respect to application requirements. Advanced studies on HP fluids have been made on the use of particular mixtures called Self-Rewetting Fluids (SRFs) [9] that increases pumping effect in the wick structure thanks to the inverse Marangoni mechanism [10]. This effect is amplified when an Inner Grooved Heat pipe (IGHP) is used instead of a porous media as wick structure as IGHP maximize the interface area between liquid and vapour where Marangoni acts.

In the literature there are available different classifications for heat exchangers with respect to transferring process, number of used fluids (examples of three-fluid heat exchangers are used in cryogenics and some chemical processes as air separation systems, helium–air separation unit, purification and liquefaction of hydrogen, ammonia gas synthesis), degrees of surface compactness, construction features, geometry, flow arrangements and heat transfer mechanisms [11]. However this manuscript will only focus on the passive, bi-phase, mono-liquid Inner Grooved Heat Pipes (IGHPs) using pure liquids and SRFs.

When IGHP is working with a non-SRF (let’s name it simply standard fluid), the liquid phase in the channel, flowing from condenser to evaporator side, will show a decrease of the radius of curvature of the liquid caused by the intrinsic meniscus receding into the corner due to the evaporator process. The liquid film gradually becomes thinner and more curved (lower radius of curvature) at the evaporator end and the difference of meniscus’ curvature between the evaporator and the condenser is proportional to the capillary pressure according to the Young-Laplace relation. When Self-Rewetting Fluids are used as working fluid, the surface tension variation along the free interface along the pipe groove (due to the presence of a thermal gradient) generates an additional flow, acting at the meniscus level, towards the evaporator. This additional flow is generated by the inverse Marangoni effect that is the result of chemical-physical properties of the mixture in specific working conditions.

Abe et al. [12] was one of the first to point out to the “beneficial effect” of Self-Rewetting Fluids for heat transfer applications; afterwards research studies have been performed to test SRFs in common wicked heat pipes [13] or pulsating heat pipes (PHP) recently [14], also in microgravity conditions [15]. The results showed an evident increase in uniformity of the thermal profile along the HP, lower thermal resistance and shift of the pipe capillary and dry-out barriers to higher temperature regimes. For all these reasons, the European Space Agency (ESA) accepted to support a space program named SELENE (SELf-rewetting fluids for ENErgy management) aiming to investigate SRFs heat transfer phenomena.

SELENE space project development is the frame of this Thesis and it consists of an experiment candidate to fly on-board the International Space Station (ISS) performing experiments in micro-gravity condition to maximize the effect of the Marangoni on pipe performances. SELENE is part of a larger program named Thermal Platform (TP1) where several experiments aiming to investigate heat transfer phenomena have been proposed and developed under the supervision of the Space Agency with the Innovative Wickless Heat Pipe Systems for Ground and Space Applications (INWIP) program.

1.2.2 Objectives

SRF in HP is a very promising research field since thermo-solute process and its potential in heat exchange application is not totally understood today. In general, the coupled effect of capillarity and inverse Marangoni flow will increase performances of a standard pipe keeping same dimensions and operational conditions. The enhancing effect on the pipe performance will modify the operational area in Figure 1.12 extending the HP limits and so preventing dry out.

The study reported in this manuscript is mainly focused on the understanding of the general and local behaviour along an experimental IGHP replacing standard working fluids with SRF. The project is developed in the frame of a long-term space program named SELENE and the research activities are aiming at providing the needed tools in terms of diagnostics and models in prevision of microgravity experiments. The resulting work will so consist mainly in four parts:

- Understanding of the inverse Marangoni nature from literature results and introduction of SELENE breadboard design: after HP capillary basic concept, from literature review it will be explained why SRFs peculiarity is manifesting in certain operational conditions in Chapter 2. Particular attention is set on the impossibility to model SRF surface tension behaviour with temperature.

- Model of the thermo-soluto-fluidodynamics for SELENE configuration: because of the complexity of the problem, will be provided an analytical model describing the heat and solutal transfer mechanisms applying in the SELENE system with particular attention to the thermic of the breadboard. A thermal model will be also introduced to simulate breadboard thermal behaviour in prevision of the experimental approach (Chapter 3).

- Investigation and development of the concentration and optical diagnostics aiming at providing composition and shape information of the liquid along the channel. In Chapters 4 and 5 these two diagnostics will be largely argued and tested experimentally. In particular, the optical diagnostic present interesting features that suggests the applicability to additional research and industrial applications.

- Experimental approach comparing performances of various working fluids: it consists of an experimental approach adapting thermal model to experimental data to determine local information between the breadboard and the liquid contained in it. Since SELENE is basically a Copper mono-grooved heat pipe, most of the heat is transferred by conduction along the channel and so, when studying heat transfer, it must be taken into account; experimental results and more will be presented in Chapter 6.