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HEAT PIPE SCIENCE AND TECHNOLOGY: A HISTORICAL REVIEW

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ABSTRACT This presentation attempts to give a historical review of heat pipe science and technology up to the present state. In a first part, a brief introduction is given into design, operation and performance limits of heat pipes and closed two-phase thermosyphons. In a second part, the historic development of closed two-phase thermosyphons and heat pipes is highlighted. It includes the history of the International Heat Pipe Conference (IHPC) series, whose 40 years anniversary we are celebrating this year. A list of overview reports and textbooks is given. A large section of this second part deals with major inventions and developments from the origins to our time. Here one can see, rather compressed and certainly not fully complete, the multitude of heat pipe/thermosyphon designs which have been developed to solve a great variety of thermal control tasks. It has also been tried to classify the numerous members of the heat pipe family in the form of a table of passive liquid-vapour heat transfer devices. The third part of the presentation deals with applications. From five of the many application fields, some selected examples are given: thermosyphons for permafrost stabilization and de-icing; heat pipe/thermosyphon heat exchangers; cooling of electric and electronic devices and components; liquid metal heat pipes for temperature calibration, material treatment and solar applications; (open and closed) two-phase thermosyphons for passive nuclear safety systems.

KEY WORDS: heat pipes, closed two-phase thermosyphons, design and operation, history of heat pipe conferences, development of heat pipe science and technology, examples of commercial applications.

1. INTRODUCTION

The history of heat pipes started in the mid-sixties. George M. Grover at the Los Alamos National Laboratory (LANL), at that time Los Alamos Scientific Laboratory (LASL), had the idea to employ the pumping action of surface tension forces for passive heat transport in an evaporation-condensation heat transfer device (which he named heat pipe) which could be of particular interest for application in space reactors, i.e. in a micro-gravity environment. He and his group built the first sodium and water heat pipes and also demonstrated for the first time the operation of water heat pipes on a satellite flight. In fact, Grover's invention was a re-invention. Already in 1942, R. S. Gaugler of General Motors Comp. had filed a patent where he suggested the use of wick structures to transport liquid in a closed tube against gravity. The application he had in mind were refrigerators. The idea was never realized, and the patent was forgotten. In 1962, the idea to employ surface tension pumping of liquids in space applications was elaborated in a rather comprehensive way by L. Trefethen from Tufts University in Medford, MA in an internal report for General Electric's Missile and Space Vehicle Department. His ideas and suggestions were not followed up, and only in 1995 this report became known to the public; it was published as an appendix to the proceedings of the 9th IHPC, Albuquerque, NM.

After publication of the first paper on heat pipes by G. M. Grover, T. P. Cotter and G. F. Ericson in the J. Appl. Phys. 35 (1964), the (re-)invention of the heat pipe became known worldwide and immediately an ever increasing interest in this topic started both in academia and industry. The emphasis was on nuclear reactors for space applications where thermionic energy conversion was the preferred electric power generation technology. Soon many terrestrial applications showed up where gravity-assisted heat pipes could be employed. In such cases, wick structures were not needed for capillary pumping, and some people talked about “wickless heat pipes” which are, in fact, closed two-phase thermosyphons.

The history of these devices goes back more than another 100 years. The first closed two-phase thermosyphon, later called Perkins tube, was invented in the mid 1830’s by Jacob Perkins, the first in a row of four generations of ingenious engineers. His Perkins tube was widely used, e.g. in baking ovens. His great-grandson Ludlow Patton Perkins, together with W. E. Buck, filed a patent on a looped thermosyphon in 1892, which was intended to be used for transference of heat over long distances and allowed to separate the vapour and liquid flows, thereby anticipating the concept of the “separate-type heat pipe” which was suggested in the 1980’s for heat exchanger applications.

A detailed discussion of these developments is given in section 3. As an introduction to the topic, it may be appropriate to start with some comments on definitions of heat pipes and closed two-phase thermosyphons, their operation and performance (section 2). Section 4 of the presentation will then present some examples of heat pipes and thermosyphons used in five selected major application fields.

2. HEAT PIPES AND CLOSED TWO-PHASE THERMOSYPHONS

2.1 The Classical Heat Pipe

Heat pipe performance and performance limits. The heat pipe (Fig. 1) is a surface tension driven liquid-vapour phase change device, in which the condensate return is accomplished through a capillary structure. It possesses a number of advantageous features, as shown in Fig. 1, which make it a highly efficient, extremely versatile heat transport element. Figure 2 shows a variety of wick structures. The simple wick structures (screen wicks, felts, sintered powder structures, (axial) grooves of various cross sections) provide both the capillary pressure for fluid circulation and the flow path for the condenser flow; so, capillary pressure and liquid flow resistance are coupled. The composite wick structures (covered grooves, annuli, slab wicks in connection with circumferential grooves, various kinds of arterial wicks (also in connection with circumferential grooves)) allow a decoupling of capillary pressure and liquid flow resistance, and thus an improved performance. An overview on working fluids and compatible structural materials, along with operating temperature ranges is given in Fig. 3.

The operation of a heat pipe is governed by the capillary pressure. At each location along the heat pipe there exists a mechanical equilibrium across the liquid-vapour interface. The respective capillary pressures at the heat pipe ends are given in Fig. 4 [1]. The difference of evaporator and condenser capillary pressures is the driving capillary pressure. For perfect wetting in the evaporator ($\Theta_e = 0$) and flooded condenser ($\Theta_c = \pi/2$), the maximum driving capillary pressure is also given in Fig. 4. Depending on the orientation of the heat pipe (operation against gravity or with gravity support), the driving pressure can be decreased or increased by the gravitational liquid pressure head. The fluid flows cause pressure drops which are given in Fig. 4. For the vapour flow, very often the radial Reynolds number is small. For $Re_{v,rad} << 1$ and laminar flow, an analytical solution of the Navier-Stokes equation is possible and the Hagen-Poiseuille equation can be used with L_{eff} as the effective length. For turbulent flow in the adiabatic section, a modification has to be made there, e.g. the Blasius equation can be used (see Fig. 4). For the other extreme case ($Re_{v,rad}$ approaching infinite), which is not shown in Fig. 4, an analytical solution is also possible. The acceleration pressure is dominant in evaporator and condenser and the respective pressure drops in evaporator and condenser are independent of flow path length and proportional to Q^2:

$$\Delta P_{v,e+c} = 2\,(1-4/\pi^2)\,Q^2/(\rho_v d_{v,h}^4\, h_{fg}^2)$$

In this case, there is a maximum pressure recovery in the condenser of nearly 40 % of the evaporator vapour pressure drop.

For the liquid flow, the Hagen-Poiseuille equation can be employed to determine the pressure drop or one can use the Darcy representation by introducing the permeability of the wick structure. The respective correlations are given in Fig. 4. For axial groove heat pipes with relatively large free liquid surface exposed to the counter-current vapour flow, an additional pressure drop of the liquid can occur in case of high vapour velocities (relative velocities between vapour and liquid) which is induced by the shearing action of the vapour flow [2, 3]. This vapour-liquid interaction must not be confused with the (droplet) entrainment from the liquid flow at high relative velocities which causes a reduction of the condensate flow to the evaporator. The shear stress induced liquid pressure drop is given in Fig. 4.

Based on a mechanical balance of these pressure drops, one of the various performance limits, the capillary (or wicking) limit can be deduced: the driving pressure (difference) plus or minus the gravitational liquid pressure head must be equal or greater than the sum of all fluid pressure drops (Fig. 4). Once the pressure drops exceed the driving pressure, not enough condensate is pumped back to the evaporator which dries out. Depending on the operating conditions, there can be a slow dry-out or a rapid burn-out, which is characterized by a fast increase of the evaporator temperature.

At very low temperatures, e.g. close to the melting point, and consequently very low vapour densities, viscous forces dominate the vapour flow. The axial heat flux is proportional to the vapour pressure difference between evaporator and condenser $p_{v,e}$-$p_{v,c}$ and comes to a limit once $p_{v,c}$ approaches zero (viscous limit). Also at low temperatures (low vapour pressures), the sonic limit can occur. At the evaporator exit, the vapour flow velocity is highest and cannot exceed sonic velocity (analogy to converging-diverging nozzle with constant gas flow). At optimum operating temperatures/vapour pressures (well above 1 bar) of the working fluid, high axial heat fluxes are possible with related high relative velocities between the counter-current vapour and liquid flows. On the one hand, the shear stress induced in the liquid flow can retard the liquid flow (flooding effect), and thus reduce the liquid supply of the evaporator. This can be accounted for with an additional liquid-vapour pressure drop. On the other hand, liquid flow instability can occur (We ~ 1), liquid droplets can be entrained into the vapour flow and thus deplete the condensate flow to the evaporator. Especially in unprotected wick structures like axial grooves, the entrainment limit can become important. The boiling limit is associated with high radial heat fluxes which may cause nucleation. The formation of bubbles can be detrimental in wick structures from which the bubbles cannot be vented into the vapour space. This is especially the case with screen wick and arterial structures. Once an insulating vapour film forms in the evaporator, rapid overheating (burn-out) occurs. Bubble formation is tolerable in open grooves as long as the critical heat flux for pool boiling is not reached.

Correlations for the capillary limit and other major performance limits are compiled in Fig. 5. The capillary limit has been presented first by Cotter [1]. In his correlation, the shear stress induced liquid pressure drop has not been considered; grooved heat pipes were not yet developed at that time. The viscous and sonic limits have been derived by Busse [4], and the boiling and entrainment limits have been presented by Marcus [5, 6]. These major performance limits are depicted qualitatively in Fig. 6 in dependence of operating temperature (T_m = melting temperature, T_{cr} = critical temperature). The region of stable heat pipe operation is within in the shaded area.

There are additional performance limitations. The vapour continuum limitation is sometimes explicitly mentioned, but it is the same as the viscous limit. The frozen start-up limitation is another intrinsic limitation which can occur before running into the viscous or sonic limit. The heat pipe may burn out because frozen liquid does not melt fast enough (or is re-solidified) to supply the evaporator with sufficient cooling liquid. The condenser capacity (or heat transfer) limitation is an extrinsic limit; it is caused by a limited cooling capacity of the heat sink. In this case the heat pipe temperature will continuously rise for constant heat input.

Comments on heat pipe modelling. In the previous section, basic equations for the various performance limits have been presented (more equations exist, especially for the entrainment and boiling limits). These equations are derived from physical analysis and have been validated through many experiments. They are a good and rather precise basis for rapid calculation of these limits. Besides knowledge of thermophysical properties of the fluids, various geometric data are needed, e.g. hydraulic diameters, permeabilities. In general there is a good data base available in textbooks and handbooks.

The standard procedure to determine the relation of heat pipe temperature and heat input (throughput) is the thermal network model. Correlations for the needed thermal resistances, heat transfer coefficients (evaporation, condensation), saturated wick conductances and contact resistances have to be employed. The thermal network model is well established. It is a good design tool, easy to work with and very flexible.

More sophisticated approaches may be needed for more detailed and multidimensional analyses. In these cases, the Navier-Stokes equations have to be solved. This method requires considerable effort. It is successfully used, especially for analysis of mini/micro heat pipes and flat heat pipes. There have also been attempts to employ CFD codes (to avoid solving the Navier-Stokes equations), however with limited success.

2.2 The Classical Closed Two-Phase Thermosyphon

Thermosyphon performance and performance limits. The closed two-phase thermosyphon (TS) is a gravity-driven device without internal wick structure (Fig. 7). It cannot operate against gravity, and for reasonable operation requires a small positive tilt (i.e. evaporator somewhat ~~above~~ condenser), e.g. around 5°. Depending on the liquid fill charge, there is pool boiling from the pool at the bottom of the TS and surface evaporation from the down flowing condensate film.

There are three performance limitations [7]. The dry-out limitation is characterized by formation of dry patches at the evaporator wall at the lower part of the evaporator or above the liquid pool (if there is any), due to insufficient fill charge. This limit can be easily avoided by keeping the fill ratio high enough, e.g. > 0.3 - 0.4. In Fig. 8, this “low-performance” limit is schematically depicted. Takuma et al. [8] showed that there is a minimum fill ratio required to accommodate a given heat input. The derived correlation between radial dry-out heat flux and fill ratio includes thermophysical fluid properties and the heated length as parameters (Fig. 9).

Another limit, which occurs at high radial heat fluxes, is the boiling limit or burn-out limit. This limit is analogous to the critical heat flux in pool boiling. The TS operates in a nucleate boiling mode (plus evaporation from the condensate film); when the radial heat flux becomes too high, dry patches occur at the evaporator wall and ultimately a stable vapour film is formed and the wall temperature shoots up (burn-out). Correlations from pool boiling can be employed. A simple burn-out correlation from Lienhard and Dhir [9] which is derived for pool boiling is given in Fig. 9. This equation describes very well the lower part of the evaporator where pool boiling exists. For the evaporation from the condensate film this may be a rather rough assumption.

The counter-current flow limitation (CCFL) is a limit for the axial heat transport (axial heat flux). This is the decisive limitation for higher fill ratios (>0.4) and for radial heat fluxes below the boiling limit, i.e. for large enough evaporator area. In Fig. 8, a typical picture for the occurrence of CCFL is shown. With increasing heat input, there is a change of flow pattern from pool boiling to churn flow to slug flow. The flow pattern strongly depends on fill ratio. In the churn flow regime there can also be non-steady (pulsating) operation by building up of a liquid plug at the condenser end which grows and after some time overcomes the holding force of surface tension and flows down as a film (e.g. in form of rivulets). Once the axial heat flux and consequently the vapour velocity is high enough, the shear stress induced in the condensate film can hold up the film and a hanging film can be formed at the evaporator exit. The liquid supply to the evaporator thus becomes insufficient or completely interrupted and burn-out occurs in the evaporator exit region.

There is a great number of CCFL correlations in the literature [10-16]; a compilation from the textbook of Reay and Kew [10] is shown in Figs. 10, 11. All correlations, except that of Golobic and Gaspersic [17, 18] are based on the Kutateladze number and have the form $q_{ax,max} = K_L$ Ku with the latent heat parameter K_L as a group of thermophysical fluid properties. They performed a comprehensive assessment of CCFL correlations and generated an own general correlation which includes, besides evaporator geometry (d_e,L_e), only thermodynamic data of the fluid (T_c,p_c), the Pitzner acentric factor ϖ and the molecular weight M, and which they claim can be used for all fluids.

The properties group K_L is plotted versus reduced temperature in Fig. 12 for a number of low and medium temperature working fluids. Based on experiments with water TS operated at low temperatures of 60 - 80 °C and R11 TS operated at 90 °C (L_e/d_e about 30 - 50), a rule of thumb formula is shown which allows a first estimate of the maximum axial heat flux. Experimental data for water at these low temperatures showed rather high values of about 1.5 to 2 kW/cm^2. If an extra-polation to higher temperatures were justified, very high axial heat fluxes would be possible, e.g. about 4.4 kW/cm^2 at 180 °C and about 6.2 kW/cm^2 at 245 °C (Fig. 12).

The ultimate performance limitation of TS is the CCFL (if the evaporator is large enough to avoid the boiling limit). Various designs have been developed to avoid the CCFL. One solution is the insertion of flow separators [19-21]. Performance improvements of 50 - 80 % could be obtained. Another solution is the looped TS ("separate type heat pipe", see section 3.5).

Comments on TS modelling. A number of empirical correlations are available to determine the performance limits. This holds especially for the CCFL. These correlations are good tools for order of magnitude calculations. One has to be careful when they are employed beyond the underlying experimental parameter range.

The thermal network model is a good standard procedure for first design calculations. Suitable heat transfer correlations for the various thermal resistances (evaporator, condenser) have to be selected. The calculation precision strongly depends on the degree to which these correlations describe the physical situation.

For more detailed and multidimensional calculations the Navier-Stokes equations are employed, along with models to describe special effects, like pulsations, geysering, hanging film. To my knowledge no comprehensive approach has been made so far. I am also not aware of any successful application of CFD codes.

3. HISTORIC DEVELOPMENT

3.1 History of Thermosyphons; Classical Patents

The history of TS is closely related to the Perkins clan (Fig. 13, left). This was a family of great inventors of thermal devices, among them closed one- and two-phase thermosyphons.

Jacobs Perkins (1766-1849) was born and lived in Massachusetts, USA. Among his 21 US and 19 UK patents was the vapour compression refrigerator (1834/35). Though the idea was older (Oliver Evans, 1805), he was the first to build one. He had many patents on generating steam for a number of processes, patents for furnaces, boilers, steam engines, apparatus for cooking, etc. He invented the closed two-phase thermosyphon, the "Perkins Tube". The related patents of 1836 are mostly on single-phase loops for applications in cooling apparatus, steam engines, furnaces, boilers. In UK patent 7059 (1836) [22] the Perkins tube was explicitly included as a means to generate steam for steam engines for boilers (e.g. in locomotives, 1863 in France), for evaporation and boiling of fluids. One of the first major applications of the Perkins tube was in bread baking ovens. Fig. 13 (right, top) shows a gas-fired baking oven with 4 baking trays employing 4x20 Perkins tubes. Typically these Perkins tubes were made of thick-walled mild steel tubes and filled to about 1/3 of the total volume with water.

Jacob Perkin's son, Angier March Perkins (1799-1881) was born in Massachusetts, USA; he moved to London in 1827. He founded the company A. M. Perkins & Son. Among his 14 patents was the single-phase hermetic tube boiler (UK patent 6145 (1831), UK patent 8311 (1839)) [23, 24]. Fig. 13 (right, bottom) shows a Perkins hermetic tube boiler, consisting of a meandering wrought iron pipe, filled with water which had an operating pressure of up to 280 bar. Such boilers were employed to produce steam for locomotives in England already in 1836.

His two sons Angier Greenleaf Perkins and Loftus Perkins were also engineers. The younger of the two sons of Loftus Perkins, Ludlow Patton Perkins (1873-1928), had various patents. The most important one in the context of heat pipes/thermosyphons is his patent, together with W. E. Buck, on "Improvement in Devices for the Diffusion or Transference of Heat" (UK patent 22272 (1892) [25]. There, as major new idea, the looped Perkins tube, was introduced. It was proposed to transport heat over long distances in relatively small tubes. The major improvement was to avoid CCFL. The looped Perkins tube is the predecessor of the "separate type heat ~~heat~~ pipe/thermosyphon" which found great attention in the 1980's. The patent comprised a number of heating and heat removal applications of the straight (slightly elevated) Perkins tube and the looped Perkins tube (Figs. 14-16). Among others these were air heating by natural and forced convection, heating of a liquid tank, and space heating with a long looped Perkins tube.

An extension and improvement of heat exchanger design with Perkins tubes was a finned Perkins tube heat exchanger with a leak-tight separation wall between hot and cold gas streams and thus without cross contamination (Fig. 17, [26]).

There is another old patent by S. W. E. Anderson on TS application for refrigeration [27]. However, this topic is related to an anti-gravity TS (see section 3.5).

3.2 History of Heat Pipes; Classical Patents

In the history of heat pipes there are three inventors. R. S. Gaugler of General Motors Comp. filed a patent in 1942 (granted in 1944) on a heat transfer device which employed a capillary structure to suck liquid against gravity [28]. Fig. 18 shows a schematic of a refrigerator (Fig. 1 of the patent), where the refrigeration space is cooled by a heat pipe, which contains a thick wick (Figs. 3, 4, 5 of the patent) which is saturated with a refrigerant. The refrigerant is evaporated, the vapour flows down to an external cooling box with crushed ice, is condensed there, and the condensate is sucked back against gravity to be evaporated again. This potential heat pipe application was never realized, and the patent was forgotten.

J. L. Trefethen of Tufts University, Medford, MA delivered in 1962 a technical report to General Electric's Missile and Space Dept. "On the Surface-Tension Pumping of Liquids or, a Possible Role of the Candlewick in Space Exploration" (Fig. 19, [29]). His ideas and suggestions were not followed up and the report became public only in 1995 as an appendix to the proceedings of the 9th IHPC in Albuquerque, NM.

In 1963, G. M. Grover of LANL (at that time LASL) independently invented the heat pipe [30]. It was only Grover and his team at LANL who built and tested for the first time heat pipes with sodium and water as working fluids, carried out the first satellite tests of water heat pipes and were at the front of liquid metal heat pipe r & d for the next decade. Fig. 20 shows part of Grover's lab notebook where he stresses the heat transfer via capillary movement of fluids and the "pumping" action of surface tension, and a photo of Grover testing a sodium heat pipe. Grover's patent (Fig. 21) shows a schematic of the operation principle of a heat pipe (Fig.1 of the patent) demonstrating the anti-g operation, along with experimental data from a sodium heat pipe (Fig. 2 of the patent).

The first publication by Grover et al. on the heat pipe principle and experimental results with screen wick sodium heat pipes appeared in 1964 in J. Appl. Phys. [31] (Fig. 22). There the new heat transfer device is named "heat pipe" and is characterized by: "Within certain limitations on the manner of use, a heat pipe may be regarded as a synergistic engineering structure which is equivalent to a material having a thermal conductivity greatly exceeding that of any known material". In Fig. 2 of the paper, we observe some excess liquid as a sump at the bottom of the vertical 30 cm long sodium heat pipe. Fig. 4 of the paper shows temperature profiles along the length of a 90 cm long slightly inclined (evaporator up) sodium heat pipe. We observe a non-condensable gas buffer (H_2) at the condenser end.

Grover was searching for an efficient heat transfer means for space applications, viz. heating and/or cooling of thermionic direct energy conversion devices in small spacecraft nuclear reactors. The temperature requirements were at a "high" level (~ 2000 K) for the cathodes and at a "low" level (~ 1000 K) for the anodes and the radiation heat sink, respectively. For the "low" temperature level, sodium was identified as suitable working fluid; for the "high" temperature level, which is beyond the range of lithium, lead and silver have been examined. In later thermionic reactor designs, heat pipe heating of thermionic converters was no longer considered.

3.3 History of Heat Pipe Meetings and Conferences

After the first heat pipe paper in 1964, this new idea was quickly taken up by various groups worldwide, and soon publications of these groups appeared in journals and at conferences. It is no surprise that heat pipe papers were first presented at thermionics conferences. Thermionic energy conversion was in the center of r & d for nuclear electric power generation for space vehicles. Thermionic converters had the prospect of higher efficiencies than thermoelectric converters (10 % vs. 5 %). The major conferences were the Int. Conf. on Electrical Power Generation and the IEEE Thermionic Specialist Conf. in the USA. As Peter Dunn recalls, it was at the 2nd Int. Conf. on Thermionic Electrical Power Generation in Stresa, 1968 where he, Grover, Busse and Groll discussed the necessity to establish an own heat pipe conference. It took 5 years till that happened. In the years between, there was an enormous growth in publications, and various national and international meetings on heat pipes took place. At one of them, the European Round Table Discussion at Euratom Ispra in 1971, Busse, Dunn and Groll decided to finally start preparations for a first International Heat Pipe Conference which should be organized by IKE, University Stuttgart. In October 1973 this happened.

The 1st IHPC started with 46 papers and 125 participants from 10 countries (Belgium, Czechoslovakia, France, Germany, Great Britain, Israel, Italy Netherlands USA, USSR) (Fig. 23). The IHPCs had a slow start, then after the 4th IHPC they took off and 1990, 1992 there were peaks in number of participants (Minsk, 1990: 205, Beijing, 1992: 210); the number of papers were also high: around 140 to 160 between 1987 and 1995. The number of participating countries also rose, from 19/18 at Minsk/Beijing to 24 at the 10th anniversary conference in Stuttgart, 1997. The number of participants, however, dropped and reached a low in Tokyo, 1999. Then there was an increase again both in number of participants and countries, with peak in Florianopolis, 2007 (270 participants from 25 countries). There was a sharp decline in Clemson, 2010 with 115 participants from 17 countries. This was certainly a negative effect of the great crisis of the global capitalistic economy which hit many countries seriously in 2008/09. There was a significant recovery (concerning our IHPC, not of the worldwide economy) till the conference in Lyon, 2012 with 170 participants from 27 countries. Now we celebrate the 17th IHPC, the 40 years Jubilee conference, with about 130 participants from 20 countries (it is a pity that due to visa problems a few colleagues could not attend).

Fig. 24 shows the cover page of the preprints of the 1st IHPC with the founding program committee members and a photo from the technical visit to IKE's labs; it shows J. Kirkpatrick of NASA ARC discussing with the conference chairman M. Groll. NASA ARC was at that time and in the following couple of years the center of innovations for heat pipe thermal control techniques for satellite applications. There exists no photo of all five founding members, but in Fig. 25, there is a photo of G. M. Grover at the 2nd IHPC in Bologna.

Besides the IHPCs, there are two other big international heat pipe oriented conferences. The International Heat Pipe Symposia which have been founded and are sponsored by the Japan Heat Pipe Association started in 1985 and had recently (July 2013) their 11th event. The IHPSs are fully dedicated to heat pipes. They are held in countries of the Pacific Rim Region. The Minsk International Seminars on Heat Pipes, Heat Pumps, Refrigerators, Power Sources started in 1993. They are organized by the Luikov Heat & Mass Transfer Institute of the Byelorussian Academy of Sciences. They are held in 2 to 3 year intervals in Minsk. The 8th event took place in 2011.

3.4 Review Reports and Textbooks

Besides a growing number of journal papers, there appeared soon after Grover et al.'s publication review reports and full-scale textbooks on heat pipes. Fig. 26 presents a list which is hopefully rather complete.

3.5 History of Heat Pipe Science and Technology

Major events and developments in heat pipe science and technology. Fig. 27 gives an overview on major events starting from J. Perkins' single and two-phase TS till the latest major invention of the oscillating or pulsating heat pipes in 1990. Some major events in "modern" heat pipe development (after Grover's invention) are discussed in more detail in this section.

Temperature Control with Heat Pipes: Shortly after the invention of the heat pipe, first tests of water heat pipes with screen wicks were carried out onboard satellites [32, 33], and cryogenic fluids were employed in heat pipes [34]. In 1969, a patent was granted to Gray for a rotating heat pipe as hollow shaft of a gas turbine compressor (Fig. 28, left), [35, 36]. Other applications on cooling of electric rotors, turbine shafts, stators of electric motors, etc. followed [37-41]. Fig. 28 (right sight) shows another type of rotating heat pipe, as flat hollow disc for potential use to cool brakes [42].

A major area of r & d in the late 1960's and 1970's was the development of heat pipe temperature control techniques for satellite electronics cooling. The first gas-buffered VCHP was proposed by Wyatt in 1965 [43]. Pioneer work was done by Turner [44], Bienert [45-47] and Marcus who wrote the first comprehensive report on theory and design of VCPHs [48]. In the early and mid-1970's, there was an enormous development of novel designs and space testing of heat pipe temperature control elements, especially VCHPs and diode heat pipes. NASA ARC was pushing the development [49-51]. There were inhouse activities at NASA, but the main task was the support of ideas, mainly from companies, and the issuing and monitoring of respective r & d contracts. Major satellite experiments were the AHPE flown on the OAO-3 satellite in 1972 [52, 53] and the ATFE flown on the ATS-F satellite in 1974 [54- 57].

Fig. 29 describes the principle of VCHPs. In a simple (non-feedback controlled) VCHP, the aim is to maintain the vapour temperature (and thereby essentially also the evaporator temperature) at a constant level, irrespective of heat input and boundary (heat sink) conditions. The standard technique is to add a non-condensable gas (NCG) buffer to the heat pipe (Fig. 30). While the vapour temperature can to a high degree be kept constant, the more important heat source temperature (i.e. temperature of the component which has to be cooled) will inevitably rise with increasing heat input due to thermal conductance and resistance between component and heat pipe. To maintain the heat source temperature constant, the increasing temperature difference (for increasing heat load) between heat source and heat pipe has to be compensated by a respective reduction of the vapour (evaporator surface) temperature. To accomplish this, a feedback control system has to be employed (Fig. 31). The standard technique is again the employment of a NCG buffer.

In Fig. 30, two examples of non-feedback controlled VCHPs with NCG buffer are shown. One is a gas controlled VCHP with cold wicked gas reservoir at the condenser end, with wick connection to the condenser. There are also VCHPs possible with non-wicked cold gas reservoir; their performance characteristics are inferior. The figure shows the vapour temperature profiles for various heat loads. Compared to an intermediate (reference) heat load, with a more or less steep vapour-gas front in the middle part of the condenser, for increasing heat load this front is pushed towards the reservoir, due to a slight increase in vapour temperature which causes an exponential increase in vapour pressure. Thereby the condenser is opened further, i.e. a larger condenser area becomes available for removing the higher heat load and thus the difference between vapour (condenser surface) temperature and sink temperature can be maintained. This means, for constant sink temperature, the vapour temperature will remain constant. On the other hand, when the heat load is decreased below the reference value, a minor vapour temperature drop and resulting vapour pressure drop allows the gas to expand towards the evaporator, thus blocking the condenser. The reduced condenser area will become just sufficient to maintain the same difference between vapour (condenser) temperature and sink temperature, and thus keep the vapour temperature constant.

The other standard control technique employs a hot wickless reservoir inside the heat pipe evaporator. In Fig. 30, a solution is shown where the gas reservoir is connected with the condenser end by a long thin connecting tube. The movement of the vapour-gas front with varying heat load is analogous to cold reservoir gas controlled VCHP.

There are further techniques to achieve VCHP behaviour. In liquid controlled VCHPs [58], there can be a liquid reservoir in a bellows at the condenser end (variable volume of reservoir) which is controlled by external gas pressure. Or: the liquid reservoir (of constant volume) contains a NCG buffer for temperature control. Liquid control is also possible as (passive or active) feedback control.

In Fig. 31, examples of active and passive feedback controlled VCHPs are shown. The most common design is the active (electrical) feedback control technique. By sensing the heat sink temperature, an electric heater at the wicked reservoir is activated for decreasing or deactivated for increasing heat input (shut down for maximum heat input). Thereby the NCG blocks the condenser to such a degree that the required vapour temperature is established to maintain the source temperature constant, viz. large temperature difference between heat source and vapour, ΔT_{HS-v}, for high heat load and small temperature difference for low heat load. The shown passive (mechanical) feedback controlled VCHP employs a non-wicked NCG reservoir in a bellows. There is a capillary tube connection, filled with a thermal control fluid of high thermal expansion coefficient, between heat source and bellows, which allows to expand (maximum power condition, condenser wide open) or contract the bellows (minimum power condition, partly closed condenser) so that again ΔT_{HS-v} is properly adjusted and T_{HS} remains constant. Other types of VCHPs are described in [59; 60-64] and listed on Fig. 42.

Besides VCHPs, heat pipe thermal diodes are the most well known thermal control devices. Whereas a TS is by nature a thermal diode, in a wicked heat pipe, there will inevitably be a heat flow reversal once heat input (heat source) and output (heat sink) are interchanged. But a heat pipe can act quite similar like a TS, though with a smaller shut-down ratio. For different wick structures in evaporator (small capillary radius of the wick pores) and condenser (large capillary radius), the reverse heat flow can be quite small for horizontal and somewhat inclined heat pipe and become zero for vertical heat pipe of sufficient length (Fig. 32).

Fig. 33 shows the two standard designs: liquid trap technique and liquid blockage technique [65, 66]. The liquid trap technique employs a wicked liquid reservoir (“trap”) at the evaporator end, without wick connection to the evaporator, but thermally well coupled to it. In the normal (forward mode) operation, the trap is empty; all liquid working fluid is in the operating heat pipe. Once heat source and heat sink are interchanged, i.e. heat input to condenser (condenser temperature above evaporator temperature), there will be at first a reverse heat pipe operation. The vapour will be condensed in both evaporator and trap. The condensate in the trap will remain there and after some time there is no longer sufficient liquid in the heat pipe and reverse heat pipe operation comes to a standstill. A minor reverse heat flow remains due to thermal conduction in heat pipe wall (and wick).

The liquid blockage technique employs a wicked liquid reservoir at the condenser end of the heat pipe, without wick connection to the condenser, but thermally well coupled to it. In the normal (forward mode) operation, the liquid reservoir is filled with excess liquid. Upon interchange of heat source and heat sink, the liquid from the reservoir is evaporated and condensed in the evaporator, where it accumulates and ultimately blocks the evaporator (and part of the adiabatic section). Once the reservoir is dry, the only reverse heat flow is by thermal conduction in heat pipe wall and (partly) saturated wick and minor cycle of evaporation-condensation between condenser and liquid plug surface.

There are further diode designs [59], e.g. the blocking orifice diode [66, 67] (a modified liquid blockage technique) or gas controlled diodes [68, 69]. The efficiency of a thermal diode is characterized by shutdown time (time from start of shutdown till steady-state reverse flow is established), shutdown energy (thermal energy deposited in evaporator during shutdown period) and shutdown ratio (ratio of forward to reverse conductance). In a well designed diode, shutdown ratios well over 1000 can be obtained.

Hybrid systems which combine VCHP and diode operation have also been developed [59; 68, 69]. Thermal diodes can also be operated as thermal switches [59; 70-73].

Two historic space applications of heat pipe thermal control techniques are shown in Fig. 34. The Ames Heat Pipe Experiment (AHPE) was flown onboard the OAO-3 satellite in 1972. It comprised a hot gas reservoir VCHP with slab wick (stainless steel-methanol). The Advanced Thermal Control Flight Experiment ATFE) was flown onboard the ATS-6 satellite in 1974. It comprised a combination of active feedback controlled VCHP, a PCM container (octadecane) as intermediate heat sink and a liquid blockage diode (axial groove heat pipe, stainless steel, ammonia). Under normal operation there is heat input to the system via the solar absorber and the diode to the PCM container as the heat sink, and the PCM melts. Once there is no solar irradiation, the VCHP extracts heat from the PCM container (the PCM solidifies) and transfers it to the VCHP condenser which is coupled to the radiator; from there the heat is radiated to space. The diode is shut down during this phase: the working fluid is evaporated from the reservoir and condensed in the diode evaporator which becomes then blocked by the condensate. With ATFE, all three thermal control elements could be successfully demonstrated.

In the 1970’s, further innovative heat pipe designs were developed, mainly under NASA ARC contracts: the electro-hydrodynamic heat pipe [74-79], osmotic heat pipe [80, 81], various artery heat pipes and priming devices [82, 83], and the inverted meniscus heat pipe [84] the principle of which is employed in the evaporators of CPLs and LHPs (see below).

Anti-gravity thermosyphons. In the first half of the 1970’s the idea of anti-gravity heat pipes (better: thermosyphons) for terrestrial applications found renewed interest (a first patent dates back to 1940 [27], especially in the UK and USA, later also in Italy and the USSR. Various designs have been proposed. Some of them are working intermittently (periodically), based on driving pressure differences (their operation is completely different from that of the also pressure gradient driven PHPs). Some are working continuously with auxiliary power supply. In Fig. 35, one example of each design is shown. The intermittently operating anti-g TS with passive pumping module (PPM) [85-89] is provided with a liquid reservoir on top which is connected with the evaporator via a float valve and a check valve. In the operating phase, vapour is generated in the evaporator and the float valve is closed. The vapour flows down to the condenser where it is condensed and gives up its latent heat. The condensate is forced upwards back to the reservoir due to the high vapour pressure in the evaporator. The pressure difference $p_v(T_e)$ - $p_v(T_c)$ must overcome the hydrostatic pressure head $\rho_l \cdot g \cdot (h_{res} - h_c)$. Once the evaporator is discharged (nearly empty), the float valve opens and the vapour pressures in evaporator and reservoir are equalized. Now the hydrostatic pressure head in the liquid line between reservoir and evaporator opens the check valve and the evaporator is refilled. Once the reservoir is empty and the check valve closed, the operating phase starts again [88]. A prototype system with R114 as working fluid was able to transport 1 kW over a height of 15 m with a temperature difference of 10 K and operating at a vapour pressure of 9 bar.

The other system is a continuously operating anti-g TS with bubble pump (vapour lift pump) [90-92]. This vapour lift pump consists of a slender riser tube inside the TS, with its lower end immersed in the condensate pool, and with an aperture at the upper end which connects the tube with the annular evaporator. The vapour lift pump contains an electric heater in the riser tube which generates vapour bubbles. In the continuous operation, vapour generated in the evaporator flows down to the condenser, is condensed there, and the condensate is pumped back by the generated slug flow in the riser tube. The required heat input is small compared to the heat transferred through the TS. Typical performance data for water as working fluid in a 20 mm diameter TS with riser tube diameter 10 mm are: axial heat fluxes of 1.2 kW/cm^2 and 4 kW/cm^2 for (lift height to pump length)-ratios a/b of 10 and 5, respectively.

Capillary pumped loops (CPL) and loop heat pipes (LHP). These types of heat pipes have had a prominent position in recent IHPCs, also in the present one. Both employ high performance wicks (inverted meniscus principle) and have perfectly separated vapour and liquid flows, so no CCFL can occur. A liquid reservoir is provided to guarantee reliable start-up and power transients. In CPLs, the reservoir is separated from the evaporator. In LHPs, the reservoir is, in general, combined with the evaporator to form an integral evaporator compensation chamber unit (Fig. 35). For start-up, liquid has to be supplied to the evaporator. This can be accomplished by active heater control of the reservoir. While this is required for CPLs, it is not mandatory for LHPs. In terrestrial LHP applications, the compensation chamber can be separated from and arranged above the evaporator, thus improving the liquid supply to the evaporator.

The concept of the CPL [93-96] is based on an idea of F. J. Stenger of NASA LeRC (1966) [93]. He introduced the concept and carried out first experimental investigations. The intention was to develop powerful heat transfer systems for space craft which would have a significantly higher performance than simple heat pipes and be a strong/superior competitor to mechanically pumped loops. Their operation would be passive, no rotary pumps would be required. In the 1970’s and 1980’s, an intense development took place, especially pushed by NASA GSFC and OAO company. First flight tests were carried out in the mid-1980’s: 1985 and 1986 onboard Space Shuttle [94-96].

The LHP [97-102] was invented in 1972 by Yuri Gerasimov and Yuri Maydanik at the Ural Polytechnical Institute [97-99]. In a first period of development, the emphasis was on high performance evaporators which would allow to pump condensate against gravity (or in micro-gravity) for high heat loads. Both terrestrial and space applications were considered. In a second period in the mid-1980’s, space applications came into the main focus. First flight experiments took place in 1989 onboard spacecraft Horizont and Granat. The first operational LHPs were employed in 1994 onboard spacecraft Obzor and 1996 onboard spacecraft Mars 8. While the initial developments took place in the USSR and were not noticed outside, the situation changed drastically after the 7th IHPC in Minsk (1990) where a number of LHPs were presented in an exhibition. The interest of the international heat pipe community was enormous and immediately the LHP technology spread worldwide. In this third period of development, the applications were strictly space oriented. LHPs were developed in Europe, USA, Japan and, after the 8th IHPC in Beijing (1992) also in China. The first European flight experiments were in 1994 (TPX/G557) and in 1998 (TPX/G467), both onboard Space Station. In a fourth period of development, starting in the early 2000’s, the focus went to terrestrial applications. Mini LHP were developed for cooling of computer and other terrestrial electronics. Efficient and cheap prototypes are developed, and their commercial production seems to be coming soon [100-104].

Typical performance data for high performance LHPs are: thermal resistance 0.05 K/W, heat transport capacity 1.5 kW/cm^2, evaporator heat flux 1 kW/cm^2, evaporator heat transfer coefficient 200 kW/m^{2}K. Respective data for mini LHPs are: cylindrical evaporators with diameters < 8 mm, flat evaporators (disk-shaped or rectangular) with thickness < 10 mm (diameters < 40 mm), vapour and liquid line diameters < 3 mm, loop length 0.5 to 1 m, heat transport capability 50 Wm, thermal resistance 0.23 – 0.5 K/W [103]. Fig. 37 [104] shows some high performance sintered metal wicks with different vapour groove arrangements. Especially fine wick structures with pore radii below 1 μm are obtained with nickel

A figure of merit for CPLs and LHPs has been suggested by Dunbar and Cadell [105] to compare working fluids: FOM = $\sigma h_{fg}^{1.75}\rho_v/\mu_v^{0.25}$. In addition, the gradient of the vapour pressure curve should be high, so that for a given difference between the temperature of the vapour in the evaporator and above the liquid-vapour interface in the reservoir or compensation chamber an appropriately high pressure difference is created to ensure displacement of the working fluid from the vapour line and condenser to the evaporator.

Comments on CPL and LHP modelling. The modelling of CPL and LHP has improved over the years. Steady-state operation can well be simulated with lumped parameter models. Still a problem is the treatment of transient situations, like start-up and power transients. Simulation of start-up is especially complicated since there can be various (not known) starting conditions. Therefore, to guarantee proper start-up, one has to ensure that the evaporator is filled with liquid. For detailed analysis of the evaporator, a multi- dimensional approach is necessary: The treatment of start-up still seems to be a field where further research is required.

Separate type heat pipes. For large-scale applications of TS in waste heat recovery systems, the "separate type heat pipe", in fact a looped TS, has been developed and widely applied in Japan and China [106, 107] (Fig. 38). In this loop solution, the flows of vapour and liquid are completely separated, and thus CCFL is avoided. This system allows to arrange evaporator and condenser far apart from each other (condenser above evaporator), e.g. in industrial production halls. These systems are significantly more efficient than conventional heat pipe heat exchangers (see section 4.2). Different from the schematic shown in Fig. 38, large systems employ in reality not one but a number of vapour and liquid lines. Thereby a certain redundancy is obtained, which of course, cannot compete with the redundancy of conventional heat pipe heat exchangers with hundreds or thousands of independent heat transfer elements. The principle of the separate type heat pipe is already included in the patent of Perkins and Buck of 1892 [25].

Micro heat pipes. At the 5th IHPC in Tsukuba (1984), Cotter [108] presented the idea of a micro heat pipe which could find applications in micro electronics cooling, e.g. by directly etching micro channels into silicon chips. Cotter defined a micro heat pipe as "so small that the mean curvature of the liquid-vapour interface is comparable in magnitude to the reciprocal of the hydraulic radius of the total flow channel", i.e. $1/r_c \sim 1/r_{h,v}$ or $r_c \sim r_{h,v}$, in conventional heat pipes we have: $r_c << r_{h,v}$. Another definition is Bo <~ 2 [109] (Fig. 39). Typical cross sections of micro heat pipes, as shown in Fig. 40, are circular, triangular or rectangular with sharp internal edges where, due to surface tension, liquid fillets can form and be pumped like in arteries.

First experimental work was strongly supported by A. Itoh who built a number of metallic prototypes (Sterling silver, copper) which were tested in various labs [110-113]. Then a number of academic and industrial groups started to develop all kinds of small flat heat pipes, often as miniaturized grooved heat pipes where the individual grooves can be treated as micro heat pipes. Other designs employed micro-machined or etched grooved surfaces in flat heat spreaders [114-116]. There have also been investigations of micro TS [117-119]. In vertically operating small diameter TS (diameter 1.5 mm), a slug flow pattern was observed with non-stationary operation and premature dry-out. By installing a small closed tube or wire inside the mini TS, the operation could be stabilized and a micro heat pipe operation with surface tension controlled condensate flow in the wedges could be established.

Cotter presented an analytical model of micro heat pipe performance and obtained an expression for the maximum performance (capillary limit) of a micro heat pipe (Fig. 40). This model under-predicted the capillary limit. Other improved analytical models [109, 111, 120] provided reasonably well agreement with experimental data, as shown in Fig. 40. There have also been a number of further investigations where the Navier-

Stokes equations were not only solved for the axial flows but also for the transverse liquid flow in the liquid wedges [121-127].

Oscillating or pulsating heat pipes (OHP, PHP). The last big invention in heat pipe science and technology are the OHP or PHP which were invented by Akachi around 1990. His major patents are from 1990, 1993 and 1996 [128-130]. These devices have found major research interest. They have opened the door to basic investigations on micro scale thermo-fluiddynamic phenomena of pulsating vapour bubble/liquid slug systems. A PHP is a most simple heat transfer device, however with very complex operational behaviour. It consists of a meandering capillary tube (or, in case of a flat plate PHP, a channel cut into the plate) and has no wick inside. It is partly filled with the working fluid. The near-optimum fill ratio is 50 % of the total volume. Due to the small diameter, the filled-in liquid distributes itself in a random way in liquid slugs and vapour plugs (elongated confined bubbles). The required diameter for surface tension dominated flow is $d_{cr} <\sim 2\ [\sigma/g\ (\rho_l-\rho_v)]^{0.5}$ or Bo $<\sim$ 2. The PHP can be designed either as an open loop (or closed end) system or as a closed loop (or open end) system: open loop PHP (OLPHP) or closed loop PHP (CLPHP). In the latter case, one or more check valves can be built in to guarantee a distinct flow direction (Fig. 41). Once there is heat input to the PHP, the generated temperature and pressure differences between evaporator and condenser induce oscillations of the liquid slug/vapour plug system. Frequency and amplitude depend on various parameters (PHP design, working fluid, orientation), notably on heat input.

Many investigations on PHP performance have been carried out so far and there is still great interest in this topic [131-141]. CLPHPs show an especially interesting behaviour. In CLPHP, the operation mode can change significantly. With increasing heat input the slug/plug system is set into motion, as indicated by zones A, B in Fig. 41. The oscillations increase and change to circulatory slug flow which reverses direction in irregular intervals. Then the flow becomes unidirectional (zone C) and slug flow changes to semi-annular and annular flow in the hot branches and bubbly flow in the cold branches (zone D). Ultimately dry-out occurs (zone E). This change in flow pattern is accompanied by a drop of thermal resistance up till dry-out. The CLPHP performs best in a non-oscillating mode. Some photos in Fig. 41 illustrate the various flow patterns.

A number of basic investigations on oscillating slug/plug flows have been carried out, with the aim to obtain an in-depth understanding of the complex thermo-fluiddynamics and to be able to predict PHP performance. More or less elaborate models have been established, but there is still no satisfying performance prediction possible [142-146]. A number of semi-empirical correlations have also been generated, however few have a solid physical basis. Moreover, their range of application is limited to the investigated experimental parameter range [137, 138]. The same limitations hold for the artificial neuronal network approach [147]. Some review articles have also been published over the years [148-152].

3.6 Systematic of Passive Liquid-Vapour Phase-Change Heat Transfer Devices

In Fig. 42, a classification of passive liquid-vapour phase-change heat transfer devices is given based on the driving force field. A distinction is made between internal and external force field driven devices. External force field driven devices comprise the following.

- Gravity driven devices are the open and closed two-phase TS, including supercritical TS and the anti-g TS with vapour lift pump.
- Centrifugal field driven devices are the rotating heat pipes.
- Electric field driven devices comprise electro-hydrodynamic and electro-osmotic heat pipes.
- Magnetic field driven devices are the magnetic fluid heat pipes. In recent years, a number of investigations have been carried out with magnetic fluids in the frame of nanofluid heat pipes. Usually suspensions of water with added ferritic nanoparticles are employed, and the heat pipe is operated in a magnetic field which shall enhance the capillary action.
- Internal force field driven devices comprise the following.
- Surface tension driven devices comprise a number of different designs: constant and variable conductance heat pipes, heat pipe diodes, heat pipe thermal switches and heat pipe triodes. There are also the CPL and LHP which are provided with highly efficient wick structures, while the micro heat pipes possess no extra wick structure but are provided with a design-inherent capillary structure.

- Pressure gradient driven devices comprise the various types of PHP (OHP) and the intermittent/periodic operating anti-g TS.
- Concentration field driven devices are the osmotic heat pipes.

The terminology in this paper is not quite in agreement with this systematic. For reasons of simplicity and also because some terms have become common use, the word heat pipe is widely used, also in cases where closed two-phase TS would be correct. I hope on the leniency of heat pipe purists.

4. HEAT PIPE / THERMOSYPHON APPLICATIONS

Heat pipes have found a great variety of commercial application fields. Major fields are:

- Permafrost stabilization in arctic zones, deicing of streets, snow removal from roofs in snow rich areas.
- Heat exchangers for waste heat recovery comprise air conditioning, cooling of electronic cabinets of industrial production machines, heat recovery in industrial processes, power stations (combustion air pre-heating, flue gas after-heating, etc.).
- Cooling of electric and electronic components in terrestrial applications and aeronautic and satellite thermal control.
- Isothermal inserts for tubular furnaces (furnace liners) are used for calibration of temperature sensors, for doping of semiconductor wafers and for annealing of small metallic parts.
- Solar energy applications: heat pipes are used in solar collectors (usually copper/water heat pipes) and in or as solar receivers (usually sodium as working fluid).
- Cooling of die-casting tools and injection moulds.
- Thermal control of chemical reactors.
- Passive emergency cooling of nuclear reactors under accident scenarios.

Some examples from selected application fields are presented in the following sections.

4.1 Permafrost Stabilization and Deicing of Streets [153-155]

The Trans Alaska Oil Pipeline was built in 1975-1977 at a cost of 8 bio US$. It connects the oil fields of Proudhoe Bay with the ice-free port of Valdez, over a length of 1285 km (Fig. 43). It can transport about 35 mio t/a in 1.22m diameter pipes. Part of the pipeline (~ 730 km) runs over permafrost soil which has to be stabilized to avoid pole jacking and possibly resulting cracks in the pipeline. For this purpose, over 130 000 carbon steel/ammonia TS were employed. For this huge one-time heat pipe project, a manufacturing factory was established and disassembled after the job was done. Fig. 44 shows a stretch of installed TS, the pipeline resting on shoe slides which can move laterally on the support beam. The latter is attached to a pole on each side (steel cylinder with 0.46 m diameter and 9 to 23 m long). Inside each pole are two TS of similar length (51 mm and 38 mm outer and inner diameter, respectively) which are provided with a finned 4 to 8 ft long aluminum cooler on top. The ammonia fill charge forms a pool at the bottom of about 1 to 2 ft depth. The function of the TS is to subcool the permafrost in winter when the air temperature is much lower than the soil temperature. In summer, when air temperatures can be well above 20°C, there is no heat pipe action (TS acts as diode), so heat can only be transported by thermal conduction into the ground. The subcooled permafrost will not melt around the pole, so the pole will not sink into the ground and then be re-lifted in winter when the molten layer freezes again. Thus, pole jacking is avoided with no mechanical stresses on pipeline.

Fig. 45 shows a rather recent similar application in China. The Qinghai-Tibet (Qingzang-Lhasa) rail-way has been built in two stretches. The first from Xining to Golmud was opened in 1984, the second from Golmud to Lhasa in 2006. This railway of 1956 km length is the highest in the world; maximum elevation is 5072 m. In some permafrost regions (length of about 500 km), two rows of mild steel/ammonia TS (total of about 10 000) are embedded along each side of the track.

Fig. 46 shows the installation of ice/snow melting TS systems on roads in the north of Japan, a snow-free road and a snow-free parking lot where deicing TS systems have been installed. Various heat sources can be and are employed: geothermal energy from the relatively warm soil °C), hot springs and also fossil-fired water boilers.

4.2 Heat Recovery

Besides satellite thermal control, heat recovery became the first commercial application field. There are many different applications and a great variety of, mostly finned, TS are used (Fig. 47). In Fig. 47, the various advantageous features of TS heat exchangers (usually called heat pipe heat exchangers, HPHX) are also briefly described. One application field is the cooling of electronic and electrical controls of production machines which are sometimes operating in the vicinity of polluted air (dust, oil aerosols, etc.). To cool the air-tight electronics cabinet, HPHX are employed. HPHX for air conditioning in commercial or public buildings usually also employ aluminium/ refrigerant TS. For application in hospitals (surgery rooms), the gas-tight partition wall is especially favourable, because cross contamination between the air flows is prevented. The same is an important aspect when HPHX are employed in power stations in connection with flue gas cleaning (see below). In big HX for heat recovery in chemical, petrochemical, pharmaceutical and other industries and in power stations, usually 1.5 to 2 inch diameter carbon steel/water TS are employed. For higher hot side temperatures (around 380 °C or higher), organic fluids like toluene or diphenyl can be used. The development of HPHX has been mainly pushed by Q-dot Corp. [156,157].

Fig. 48 shows one of the early large scale HPHX applications [158]. It employs HPHX in an incineration plant in Vienna. The flue gas from two boilers undergoes first desulphurization. Then the cool gas has to be heated up to the operation temperature of the selective catalytic reactor (SCR) (~ 300 °C). To minimize the heating energy, a HPHX is installed in front of the SCR. There the cold flue gas is heated up from ~ 85 °C to ~ 220 °C, then brought to ~ 300 °C via an afterburner. In the lower exit part of the HPHX the clean flue gas is cooled down from ~ 280 °C to ~ 150 °C, before entering another HX system where district heating water is heated up. Then the cold clean flue gas (~100 °C to 120 °C) enters the chimney. The HPHX contains 3000 TS, each 10.5 m long. The TS are arranged in modules, as shown in Fig. 47. The gas streams from the two boilers are 240 000 Nm^3/h, the heat duty is 19.2 MW.

A blast furnace application is shown in Fig. 49. Both combustion air and combustion gas (low caloric value blast furnace gas) are preheated with flue gas from the cowper which then enters the chimney with about 120 °C. In this application, the flow rates are about 145 000 Nm^3/h for air preheating (heat duty 12.6 MW) and about 200 000 Nm^3/h for gas preheating (heat duty 17.9 MW).

4.3 Cooling of Electric Devices and Electronic Components

A niche market for heat pipe technology is the cooling of electric motors. Among the various possibilities [7, 10, 40, 159-162], rotor cooling via a hollow shaft (as in Gray’s 1969 patent) is one straight-forward solution. Fig. 50 shows two examples of rotor shaft cooling. The upper part shows the schematic of the rotor with heat pipe shaft of a 75 kW induction motor**;** the performance increase for same superheat of the insulation of the stator windings was 17 %. The lower part shows the photo of the rotor with heat pipe shaft of a 138 kW DC motor. In Fig. 51, a number of heat pipes for cooling of electronic components, e.g. CPUs in laptops are shown, partly with attached heat sinks. In general, such heat pipes are copper/water heat pipes of circular or flattened shape with outer diameters of 2 to 8 mm for micro electronics cooling and about 8 to 15 mm for power electronics cooling. Annually about 100 mio of these heat pipes are produced. In Fig. 52, an example of laptop CPU cooling is shown with two flattened heat pipes which are attached to the same fan-cooled heat sink. Another example shows the cooling of 8 transistors with one relatively big heat pipe. The other two examples show thyristor cooling. In one, two aluminium blocks with two copper/water heat pipes each are employed. A number of such thyristors are stacked in a rectifier pile for electric resistance brakes in electric locomotives. In the other example, two cooling blocks are employed with three heat pipes each which are provided with ceramic electric insulation parts. A dielectric working fluid (e.g. FC 72) is used in this case. Such electrically insulated heat pipe coolers are employed in traction drives.

Flat mini/micro heat pipes are used in electronic packages. 3D electronic packages are attractive because they can be more densely packed than 2D packages. However there are some limitations. Existing 3D packages have no sufficient power dissipation capacity, they allow no flexible assembly technique (there is a fix block sealed with epoxy). And consequently there is no modularity and interchangeability of slices. So the idea came up to develop in the frame of a multinational EU project a design characterized by following features: stacked structure with 3 layers, total thermal power dissipation 30 W (10 W per layer), possibility to integrate slices of different technologies, modularity and standardized interchangeability of slices, adaptability to liquid cooling, minimum possible volume and weight, and increase of reliability. The development was carried out by 5 companies (Thales (FR) as project coordinator, Alcatel/Space (FR), Nokia (FI), Electrovac (AT), Customs Interconnect (GB)) and 2 university institutes (INSA Lyon (FR) and IKE; University Stuttgart (DE)). The project was carried out from 2000 to 2003. Fig. 53 shows a summary of the results. Demonstrator slices were developed by Nokia for terrestrial, by Thales for avionics and by Alcatel/Space for space applications. The employed flat mini/micro heat pipe is made of AlSiC plates with overall dimensions 40x50x0.9 mm^3. The capillary structure is made of two layered 325 mesh CuSn screens sintered to the heat pipe bottom plate. There is about 0.25 mm vapour space thickness left. Working fluid is water. The cross section of the package shows the solution with finned heat sink for convective air cooling. An option is to include one or two small heat pipes with 2 to 3 mm outer diameter in each of the 3 sections of the thermal path. The exploded view indicates that 2 sides of the stack are used for electric connection and 2 sides for heat removal to the heat sink [163 - 165].

The present development of mini LHP for terrestrial electronics cooling of computers is characterized by copper/water or stainless steel/ammonia systems with powerful evaporator wicks. The designs presented in Fig. 54 are (clockwise): two Cu/H_2O systems with flat evaporator (3 mm thick), power range 5-160 W; three StSt/NH_3 systems with cylindrical evaporator (diameter 5 to 6 mm), power range 5-50 W; one StSt/NH_3 system with cylindrical evaporator (diameter 8 mm), power range 5-120 W, 2 m long flexible fluid line; three StSt/NH_3 systems with flat evaporator (10 to 13 mm thick, diameter 30 mm), power range 10 to 100 W. Fluid lines have a diameter of 2 mm [104].

In another EU project (2005-2008), a LHP cooling system was developed for the seat electronic box (SEB) which is managing the in-flight entertainment (IFE) in long-distance commercial aircraft. The project companies were Thales (FR) as project coordinator, EHP (BE), Recaro (DE) and the academic institutions ITP, Ural Branch of Russian Academy of Sciences, INSA Lyon (FR) and IKE, University Stuttgart (DE) [166]. The cooling system comprised two mini LHP (Cu/R141b, power capacity 50 W per LHP). Fig. 55 shows the two mini LHP, with the evaporators attached to two opposite sides of the SEB, and the condensers attached to the seat frame as heat sink [104].

Pulsating heat pipes have found applications in various electronic cooling tasks. Fig. 56 shows a number of examples: five forced convection air cooled Kenzan Fins with square base plates, one with a heat transport capability of 450 W, and one modified version, the Stereo-Type Heat Lane Heat Sink and a Kenzan Fin design as IGBT cooler.

4.4 Liquid Metal Heat Pipes

Liquid metals are used as working fluids in various niche applications. Fig. 57 shows two coaxial heat pipes with different black body cavities which are used as isothermal furnace liners for tubular furnaces. This combination of heat pipe with standard tubular furnace provides an excellent tool for calibration duties. Usual cavity emissivities are greater than 0.999. The heat pipe with the big cavity serves for pyrometer calibration; the small cavity contains a control thermometer. The multi-port black body is used for simultaneous calibration of different thermometers (thermistors, thermocouples) [167]. Such calibration devices can be employed in a wide temperature range, from about -50 °C (NH_3 as working fluid) up to 1100 °C (Na as working fluid). In the temperature range between, H_2O and Cs are used. For even higher temperatures, up to about 1600 °C, Li is used.

The Au-fixpoint black body is an excellent cost effective calibration device with very good isothermality (+- 0.05 K) and temperature stability (+- 0.1 K). The photo shows the solidified chunk of gold with the aperture end of the graphite crucible. The latter is inserted in a super alloy/sodium coaxial heat pipe. In the photo of the complete system (upper part: box for black body, heating system and thermal insulation; lower part: box for electronics, heater control, etc.) one can see the end of the coaxial heat pipe protruding out of the upper box.

Besides calibration tasks, coaxial liquid metal heat pipes have also been developed for doping of Si wafers and annealing of small metallic parts.

Liquid metals (mostly Na) have also been employed in high temperature solar power applications [167-169]. Fig. 58 shows a Na heat pipe receiver for the heater head of a V-type Stirling engine. The cavity receiver (inner diameter 180 mm, 270 mm length) consists of a coaxial heat pipe with a screen wick capillary structure; one end of the heat pipe is closed with a slightly conical ceramic part. The lower half of the outer heat pipe envelope is provided with helical grooves into which the He tubes of the Stirling engine heater head are fixed. The photo shows the complete Stirling engine-generator unit mounted in the focal spot of a 10 kW_{el} solar dish power station. About 950 hours of on-sun tests have been accumulated. The He temperatures were about 700 °C. Advantages of the heat pipe solution are more uniform heating of the He tubes and smoother operation of the Stirling engine, and improved system efficiency by about 10 %.

4.5 Emergency Cooling of Nuclear Reactors

In the context of the new generation of nuclear reactors (Gen 3 and 3^+), inherently safe passive systems for emergency cooling, i.e. after reactor shutdown as a consequence of a severe accident (loss of coolant accident (LOCA) with brake of major coolant line) have become a high-priority topic, and such systems have been installed in all Gen $3/3^+$ reactors. All these passive devices operate automatically in case of a LOCA and allow to remove the decay heat from the shut down reactor without any auxiliary power. The idea of a heat pipe cooled core catcher is very old; it has already been discussed in the mid-1960’s. In the 1980’s, many studies have been performed worldwide [e.g. 170-172].

Fig. 59 shows a schematic of a pressurized water reactor with various heat removal systems which employ standard TS and looped TS (separate type heat pipes). The standard TS are employed for core catcher cooling and cooling of the wall of the reactor pit (system HP3 in Fig. 59, left side; elements 1a,b and 5,6 in Fig. 59, right side). These are carbon steel/water TS, the heat sink is an external water pond (18). For containment cooling a “loop heat pipe HX” (separate type heat pipe HX, closed two-phase TS) is employed (system HP1 in Fig. 59, left side). Two similar systems (HP2 and HP4) serve for internal cooling of the reactor pressure vessel and for cooling of the reactor cavity. Heat sink for HP1 is a roof top water reservoir (17). Flooding container and storage tank IRWST serve as heat sinks for HP2 and HP4.

Two emergency cooling solutions for the ESBWR are shown in Figs. 60 and 61, viz. the passive containment cooling system (PCCS) and the isolation condenser system (ICS). The ESBWR is a further development of the ABWR, of which 11 systems have been built in Japan and two in the USA [173]. The ESBWR is in the final stage of certification. The pressurized water reactors AP 1000 and VVER 1000 have similar safety features. A number of VVER reactors are already operating in Russia and China, two have been built in India (the second one has been recently connected to the grid). The AP 1000 has been built in China, power stations are planned in USA and India. The two major emergency cooling systems (PCCS and ICS) have the following characteristics.

ICS: After a steam line rupture, high pressure steam from the RPV is released to the isolation condensers inside the PCCS pool. The condensate flows back to the RPV and is re-evaporated. This system acts as closed looped two-phase TS. After the high pressure cooling phase, the steam relief valves (SRV) are automatically activated to depressurize the RPV. Then the released steam from the RPV is injected into the suppression pool via a direct connecting line and a nozzle system. In the following low pressure cooling phase, water from the GDCS pool flows into the RPV driven by gravity.

PCCS: After steam line rupture, the released steam fills the containment. To avoid unacceptable pressure build-up, steam is guided to both the suppression pool and the PCCS HX inside the PCCS pool where it is condensed. The condensate flows into the GDCS pool. As soon as the RPV pressure is low enough (< 1 bar), water from the GDCS pool can flow into the RPV, driven by gravity, and is evaporated again. The PCCS/GDCS cycle acts as open two-phase TS.

5. SUMMARY AND CONCLUSIONS

The gravity driven closed two-phase thermosyphons have been introduced and widely used already well over 150 years ago (Perkins tubes). In our time, they are mainly employed in the field of heat exchange and heat recovery between/from gas and liquid flows. Open two-phase thermosyphons have found a small but important niche application in passive nuclear reactor safety of the new Generation 3/3+ reactors. The surface tension driven heat pipe has gained worldwide attention after its (re-)invention by Grover in 1963. From that time on the interest in the “new” heat pipe and "old" thermosyphon grew tremendously. It has been and is strongly supported by the soon established International Heat Pipe Conference series and other following international meetings.

A multitude of heat pipe (also thermosyphon) designs have been developed for a great number of applications. But the good is sometimes the enemy of the better. There have been many cases (I know it from own experience) where superior heat transfer solutions with heat pipes have pushed engineers to improve and optimize existing systems so that the "better" heat pipe solution was put aside. However, heat pipes have conquered some application fields. This holds especially for thermal management duties in satellites. Satellite operation without heat pipes is no longer imaginable. The by far largest application (in numbers and money-wise) are copper/water heat pipes for cooling of (micro) electronic devices, especially in computers. These are mostly small cylindrical, flattened or thin flat plate heat pipes. For high performance applications, LHP and mini LHP are on the verge of becoming commercialized.

One may think that after nearly 50 years of modern heat pipe science and technology the theoretical basis should be not only broad and solid, but there should also be widely applicable design tools available. This is, however, only partly the case. "Simple" heat pipes, including flat plate and mini/micro heat pipes, can be reasonably well calculated, though partly with substantial effort. It is not surprising that the more complex phenomena in CPL, LHP and especially PHP are not easy to handle. But even performance prediction of thermosyphons is still an open field. A lot of experimental data are available, and many (semi-)empirical correlations for heat transfer and for performance limits have been generated. However, they are in general only applicable in the narrow range of the underlying experimental conditions. A comprehensive critical assessment of the piled up knowledge would be a tedious but commendable task.

For simulation of CPL and LHP performance, a lot of theoretical work has been performed. While models for steady state calculations seem to work well, the transient operation needs more attention. Here, as should be the usual case in engineering science, a close interaction of experimental and theoretical research is needed.

The most challenging task in heat pipe theory is the treatment of the youngest child in the heat pipe family, the PHP or OHP. Though grown up by now, the PHP's performance can still not be predicted. It seems that the extremely complex quasi-chaotic behaviour, coupled with various operation modes (especially in CLPHP) makes their modelling nearly impossible. Application of chaos theory and elementary use of conservation equations have largely failed. Some limited success could be attributed to semi-empirical correlations for the maximum heat transport capability which are well based on the physics of PHP. Their accuracy and range of application are however limited. Since a few years, activities are under way where the very basic phenomena in PHP are experimentally investigated, viz. pulsating flow of bubbles/bubble trains in adiabatic and heated micro/mini channels. Results from studies of flow boiling in micro/ mini channels can probably be adopted. In parallel refined theoretical modelling of the micro scale phenomena is carried out. But it will certainly be a long way till PHP performance can be well predicted.

In summary one can say: There is for sure enough research and development work to be done in the future. This will certainly provide very valuable results and guarantee interesting coming IHPCs.

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NOMENCLATURE

A	cross section (m^2)
b	tortuosity factor (-)
b*	width of axial groove ribs (m)
D, d	diameter (m)
E_{sd}	shutdown energy (J)
F	fill ratio, $F = V_L/V_E$ (-)
g	gravitational constant (m/s^2)
h_{fg}, Δh_{lv}	latent heat of vaporization (J/kg)
K	permeability (m^2)
K	thermal conductance (Fig. 32) (W/K)
K_L	latent heat parameter, (W/m^2)
K_L	$= h_{fg}\,[\rho_v^2\,(\sigma\, g\,(\rho_l - \rho_v)]^{0.25}$

k	thermal conductivity (W/(m*K))
L	length (m)
L	latent heat of vaporization (Figs.9, 12) (J/kg)
M	molecular weight (in Fig. 11) (kg/mol)
P, p	pressure (Pa)
p_r	reduced pressure, $p_r = p/p_{cr}$ (-)
Q	heat flow, thermal power (W)
q	heat flux (W/m^2)
R	radius (m)
R_{sd}	shutdown ratio, $R_{sd} = K_f / K_r$ (-)
R*	universal gas constant (J/(mol*K)
T	temperature (°C)
T_r	reduced temperature, $T_r = T/T_{cr}$ (-)
t	time (s)
V	volume (m^3)
v	velocity (m/s)
w	groove width (m)
Z	characteristic length (e.g. screen mesh width, groove width) (m)
z	axial coordinate (m)

Greek alphabets

α inclination angle (towards horizontal) (°)

β dimensionless geometry factor for micro heat pipes, $\beta^2 = A_l/r_{cap}(z)^2$ (-)

γ surface tension (Figs. 9, 12) (N/m)

δ groove depth, nucleation site radius (m)

η dynamic viscosity (Fig. 9) (Pa*s)

μ dynamic viscosity (Pa*s)

ν kinematic viscosity, $\nu = \mu/\rho$ (m^2/s)

ρ density (kg/m^3)

σ surface tension (N/m)

$\tau = 1 - T_r$ (Fig. 10b) (-)

ω Pitzner acentric factor (Fig. 11), (-)

$\omega = -\lg(p_s/p_{cr})_{Tr = 0.7}$

Θ wetting angle (°)

Subscripts

ad	adiabatic
ax	axial
B	boiling limit
c	condenser
c	critical (Fig. 11)
c	capillary limit (Fig. 5)
cap	capillary
cr	critical
e, E	evaporator
E	entrainment limit (Fig. 5)
eff	effective
f	forward (Fig. 32)
gra	gravitational
h,	hyd hydraulic
HS	heat source
i	inner
l, liq	liquid
m	melting
max	maximum
o	evaporator exit (Fig. 5)
r	reverse (Fig. 32)
rad	radial
res	reservoir
s, sat	saturation
s	sonic limit (Fig. 5)
sd	shutdown
v	viscous limit (Fig. 5)
v, vap	vapor

Non-dimensional groups

Bo	Bond number, $Bo=d[g(\rho_l-\rho_v)/\sigma]^{0.5}$
Ku	Kutateladze number,
Ku	$= h_{fg}\, \rho_v^{0.5}[\sigma g(\rho_l-\rho_v)]^{0.25} = q_{co}/K_L$
Re	Reynolds number, $Re = \rho v d_h/\mu$
We	Weber number, $We = \rho_v\, v^2\, Z/(2\pi\sigma)$

Abbreviations

ABWR: Advanced Boiling Water Reactor

AHPE: Ames Heat Pipe Experiment

AIAA: American Institute of Astronautics and Aeronautics

ASME: American Society of Mechanical Engineers

ATE: Applied Thermal Engineering

ATFE: Advanced Thermal Control Flight Experiment

ATS: Advanced Technology Satellite

CCFL: Counter-Current Flow Limitation

CPU: Central Processor Unit

ESBWR: Economic Simplified Boiling Water Reactor

EU: European Union

GDCS: Gravity Driven Cooling System

HPHX: Heat Pipe (Thermosyphon) Heat Exchanger

HX: Heat Exchanger

ICS: Isolation Condenser System

ICTEPG: International Conference on Thermionic Electrical Power Generation

IECEC: Intersociety Energy Conversion Engineering Conference

IEEE TSC: IEEE Thermionic Specialist Conference

IGBT: Insulated-Gate Bipolar Transistor

IHPC: International Heat Pipe Conference

IHPS: International Heat Pipe Symposium

IJHMT: International Journal of Heat and Mass Transfer

IKE: Institut fuer Kernenergetik & Energiesysteme (Institute for Nuclear Technology & Energy Systems), University Stuttgart

ITP: Institute of Thermal Physics, Ural Branch of Russian Academy of Sciences

LANL: Los Alamos National Laboratory

LASL: Los Alamos Scientific Laboratory

NASA: National Aeronautics and Space Administration

NASA ARC: NASA Ames Research Center

NASA GSFC: NASA Goddard Space Flight Center

NASA LeRC: NASA Lewis Research Center (now: NASA Glenn Research Center)

NCG: Non-Condensable Gas

OAO: Orbiting Astronomical Observatory

OHP: Oscillating Heat Pipe

PCCS: Passive Containment Cooling System

PCM: Phase-Change Material

PHP: Pulsating Heat Pipe

RPV: Reactor Pressure Vessel

SRV: Steam Relief Valve

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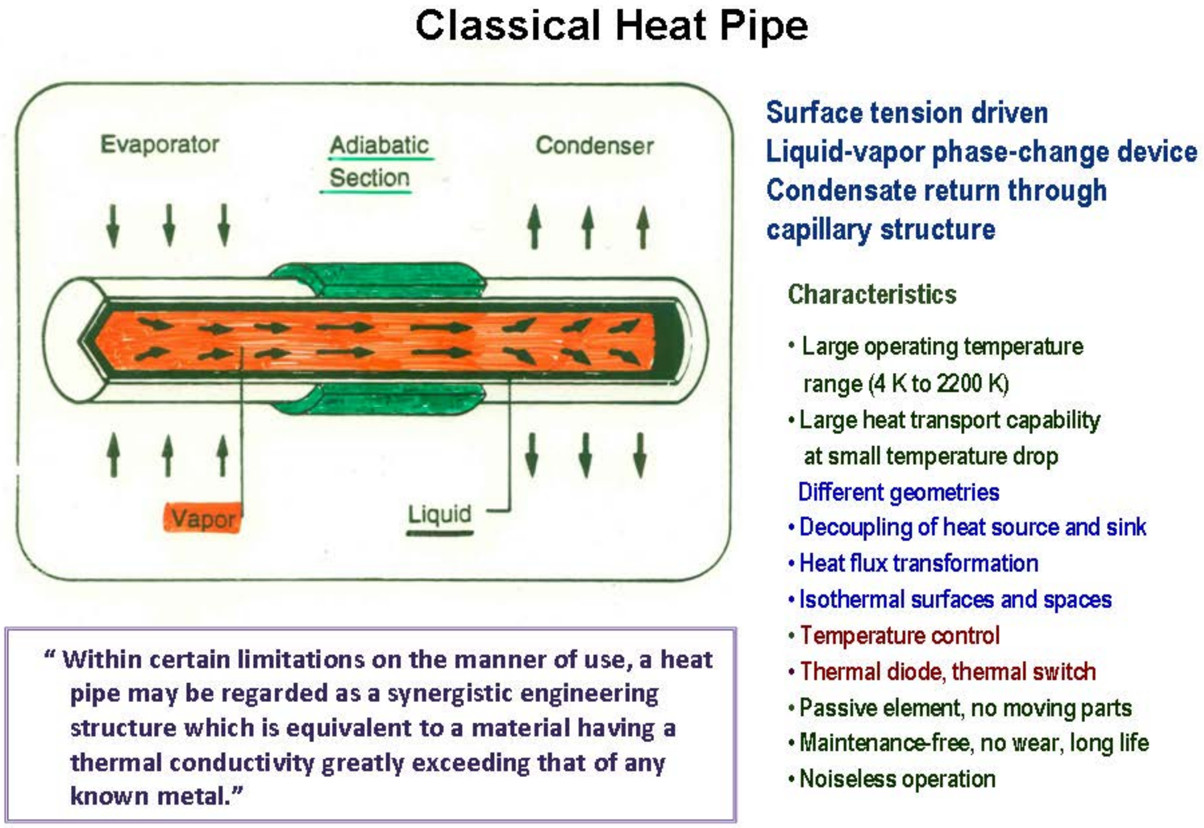


Fig. 1: Heat pipe: schematic, operation principle, characteristics

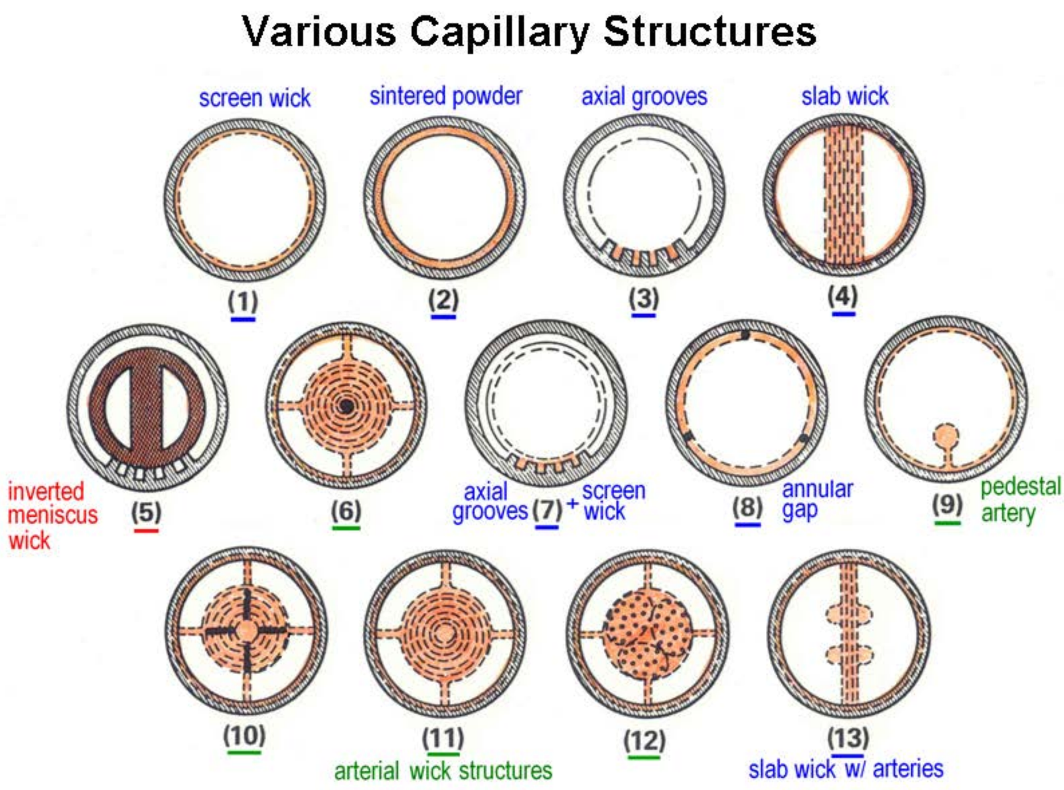


Fig. 2: Heat pipe: simple wick structures (1-3) and composite wick structures (4-13)

Working Fluids, Temperature Ranges, Structural Materials

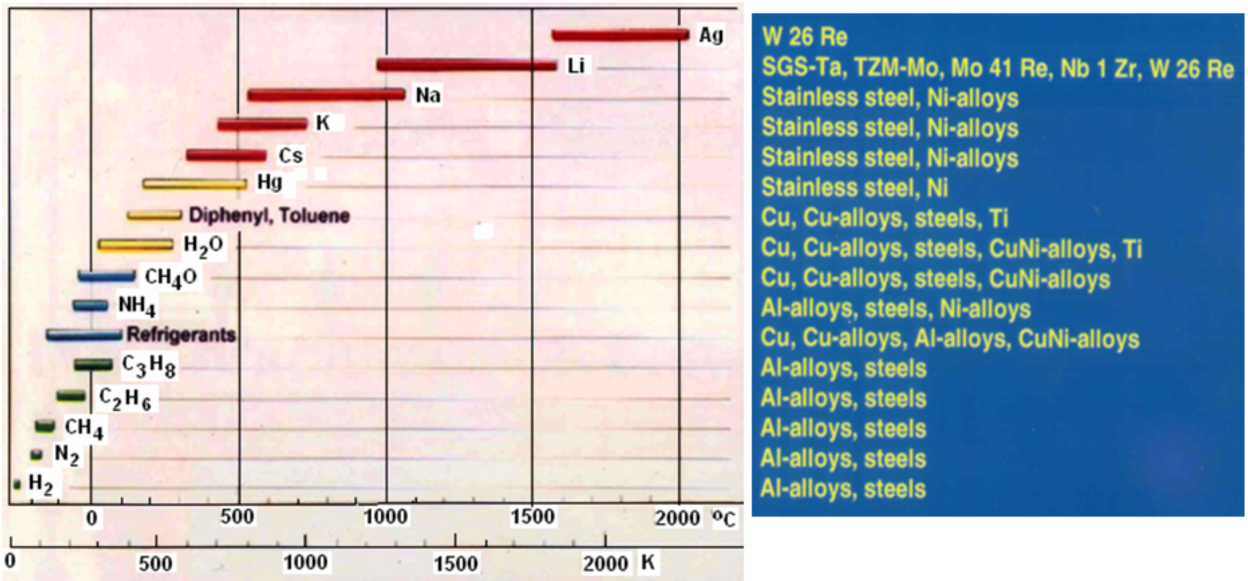
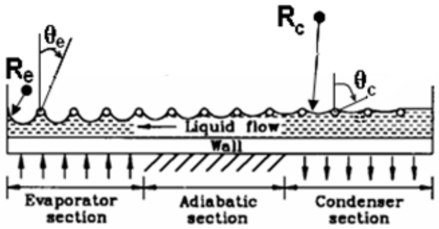


Fig. 3: Heat pipe: working fluids, operating temperature ranges, structural materials

Basic Heat Pipe Equations

Capillary Pressure



$$\Delta P_{cap,e} = \frac{2\sigma}{R_e} = \frac{2\sigma\cos\theta_e}{r_{cap}}$$

$$\Delta P_{cap,c} = \frac{2\sigma}{R_c} = \frac{2\sigma\cos\theta_c}{r_{cap}}$$

$$\Delta P_{cap} = \frac{2\sigma}{r_{cap}}(\cos\theta_e - \cos\theta_c)$$

$$\Delta P_{cap,max} = \frac{2\sigma}{r_{cap}}$$

Pressure Drops
(hydrostatic and viscous)

$$\Delta P_{gra} = \rho_l g\,(L_{tot}\sin\alpha - D\cos\alpha)$$

$$\Delta P_{liq} = \frac{b\cdot 32\cdot \mu_l \cdot L_{eff}\cdot Q}{\rho_l \cdot h_{fg}\cdot A_l \cdot D_l^2} = \frac{\mu_l \cdot Q\cdot L_{eff}}{\rho_l \cdot h_{fg}\cdot K_l \cdot A_l}$$

$$\Delta P_{vap,lam} = \frac{32\cdot \mu_v \cdot L_{eff}\cdot Q}{\rho_v \cdot h_{fg}\cdot A_v \cdot D_v^2}\ ;\ \mathrm{Re}_{ax} < 2300\ ;\ \mathrm{Re}_{ax} = \frac{Q\cdot D_{v,hyd}}{h_{fg}\cdot \mu_v \cdot A_v}$$

$$\Delta P_{v,turb} = \Delta P_{v,lam}\left(0.004875\cdot R_{ax}^{0.75}\right)\ ;\ \mathrm{Re}_{ax} > 2300$$

$$\Delta P_{liq\ vap} = \Delta P_{vap}\left(\frac{w^3\cdot A_v}{(12\cdot (b^{*}+w)\cdot K_l\cdot A_l}\right)$$

Capillary or Wicking Limit

$$\boxed{\Delta P_{cap} \pm \Delta P_{gra} \geq \Delta P_{liq} + \Delta P_{vap} + \Delta P_{vap\ liq}}$$

Fig. 4: Basic heat pipe equations

Heat Pipe Performance Limits

- **Capillary Limit**
  (Cotter, 1965)

$$q_c = \frac{h_{fg}}{32 L_{eff} A_v} \frac{\Delta p_{cap,\max} \pm \Delta p_{gra} \boxed{- \Delta p_{liq\ vap}}}{\frac{b \nu_l}{A_l d_{l,h}^2} + \frac{\nu_v}{A_v d_{v,h}^2}}$$

- **Viscous Limit**
  (Busse, 1973)

$$q_v = \frac{d_v^2 h_{fg}}{64 \rho_v \nu_v L_{eff}} \rho_{v,o} P_{v,o}$$

- **Sonic Limit**
  (Busse, 1973)

$$q_s = (0.474) h_{fg} \sqrt{\rho_{v,o} P_{v,o}}$$

- **Entrainment Limit**
  (Marcus, 1972)

$$q_E = h_{fg} \sqrt{\frac{\rho_v \sigma}{Z}}$$

- **Boiling Limit**
  (Marcus, 1972)

$$q_B = \frac{2 k_{eff,w} \Delta T_{cr}}{d_i \ln(\frac{d_i}{d_v})}$$

$$\Delta T_{cr} = \frac{3.06 \sigma T_s}{\rho_v h_{fg} \delta}$$

(Hsu, 1962)

Fig. 5: Heat pipe performance limits: classical correlations

Heat Pipe Performance Limits

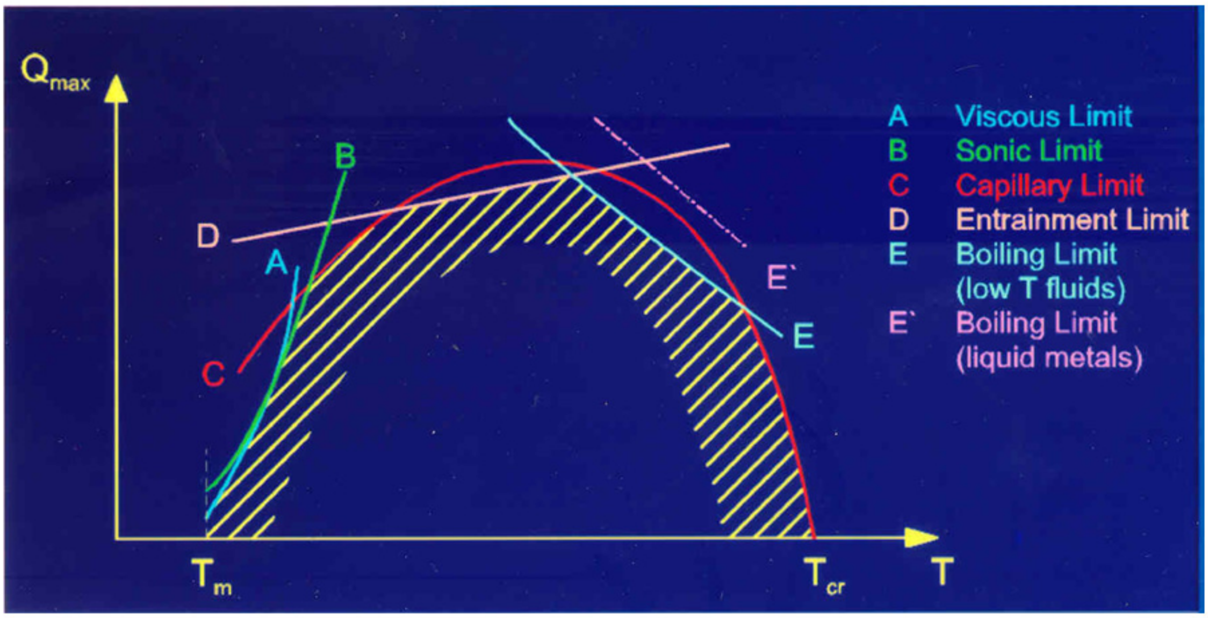


Fig. 6: Heat pipe performance limits: qualitative graphical representation

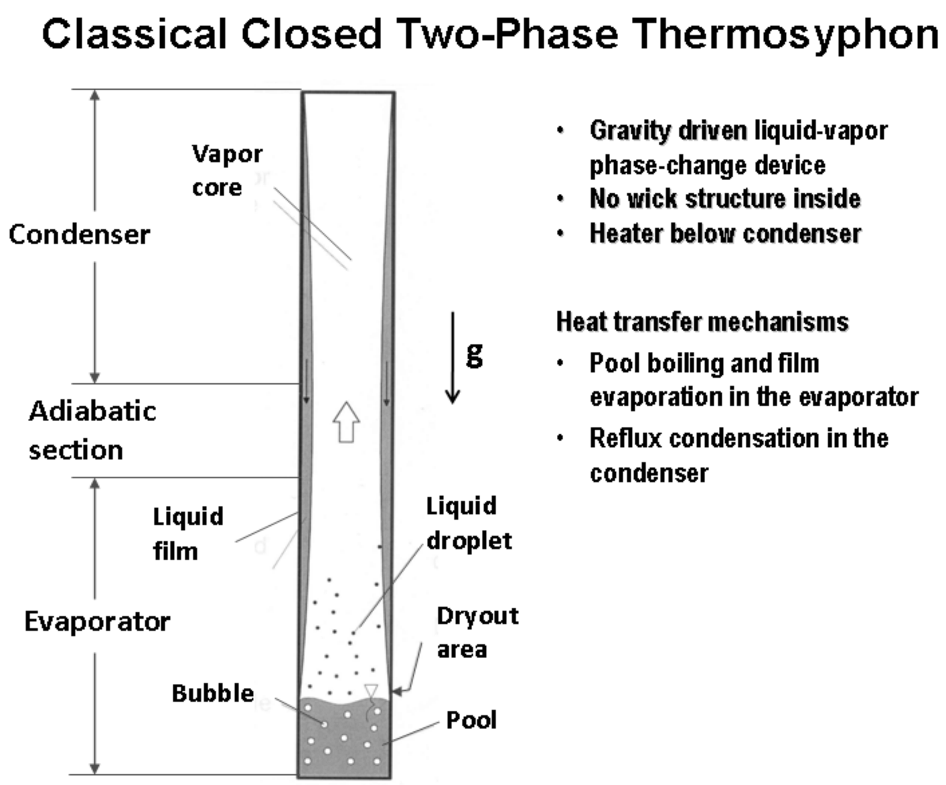


Fig. 7: Closed two-phase TS: schematic, operation principle, heat transfer mechanisms

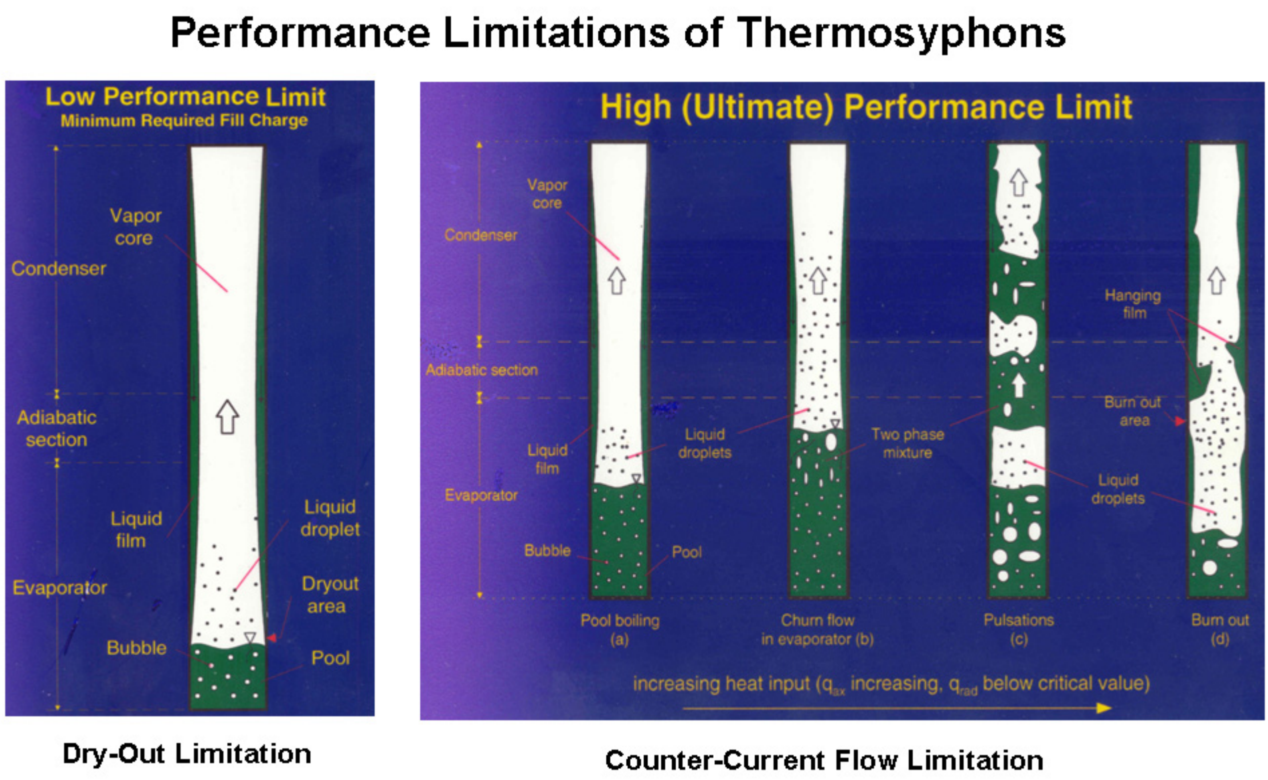


Fig. 8: Closed two-phase TS: schematics of dry-out limitation (left) and CCFL (right)

Performance Limitations of Thermosyphons

Dry-out limitation

- Small fill ratios $F=V_L/V_E$, small heat fluxes
- Due to scarcity of the liquid pool at
- bottom of evaporator
- Formation of dry patches are observed at
- evaporator wall
- This limit can easily be avoided by keeping
- FR > 0.4

$$F = 1 - \left[1 + c\left\{\left[\frac{\left(q_{\max,\,rad} / L^2\right)}{(2\gamma\rho_V)}\right]\left[\frac{\left(3q_{\max,\,rad}\,\eta_L l_E\right)}{\rho_L(\rho_L-\rho_V)gL}\right]^{\frac{1}{3}}\right\}^{\frac{3}{4}}\right]^{-1}$$

M. Takuma et al. (1986)

Boiling or burn-out limitation

- Important in case of high radial heat fluxes
- Stable film of vapor forms at evaporator walls
- Depends on geometry of heated surface and thermo-physical properties of the liquid

$$q_{\max,\,rad} \equiv \frac{Q_{\max}}{A'_E} = 0.12\,\underbrace{\boxed{L(\rho_V)^{\frac{1}{2}}\left[\gamma g(\rho_L-\rho_V)\right]^{\frac{1}{4}}}}_{\text{Latent Heat Parameter } K_L}$$

$$A'_E = \pi d_E l_E$$

J. H. Lienhard, V. K. Dhir (1973)

Counter-current flow limitation

- Limit of axial heat transfer
- Decisive for higher fill ratios (FR > 0.4) and radial heat fluxes below boiling limit (i.e. relatively large evaporator area)

$$q_{\max,\,ax} \equiv \frac{Q_{\max}}{A} = Ku\,\underbrace{\boxed{L(\rho_V)^{\frac{1}{2}}\left[\gamma g(\rho_L-\rho_V)\right]^{\frac{1}{4}}}}_{\text{Latent Heat Parameter } K_L}$$

$$A = \pi d^2/4$$

$$Ku = f_1 f_2 f_3 (d_e/L_e)^{0.14}$$

M. Groll, S. Roesler (1992)

Fig. 9: Closed two-phase TS: three performance limitations

Reference	Correlations Kutateladze number $\mathrm{Ku} = q_{\mathrm{ax}}/\underbrace{\boxed{\{\Delta h_{\mathrm{lv}}\rho_{\mathrm{v}}^{0.5}[\sigma g(\rho_{\mathrm{l}}-\rho_{\mathrm{v}})]^{0.25}\}}}_{\text{Latent Heat Parameter } K_L}$
Sakhuja [84]	$\mathrm{Ku} = \frac{0.725^2}{4} Bo^{1/2} \frac{(d_e/l_e)}{[1+(\rho_v/\rho_l)^{1/4}]^2}$
Nejat [85]	$\mathrm{Ku} = 0.09 Bo^{1/2} \frac{(d_e/l_e)^{0.9}}{[1+(\rho_v/\rho_l)^{1/4}]^2}$
Katto [86]	$\mathrm{Ku} = \frac{0.1}{1+0.491(l_e/d_e)Bo^{-0.3}}$
Tien and Chang [87]	$\mathrm{Ku} = \frac{3.2(d_e/l_e)}{4[1+(\rho_v/\rho_l)^{1/4}]^2}$; $Bo \geq 30$ $\mathrm{Ku} = \frac{3.2}{4}[\tanh(0.5Bo^{1/2})]^2 \frac{(d_e/l_e)}{[1+(\rho_v/\rho_l)^{1/4}]^2}$
Harada et al. [88]	$\mathrm{Ku} = 9.64(d_e/l_e)\rho_v h_{lv} C\left(\frac{\sigma}{\rho_v}\right)^{1/2}$; $C\left(\frac{\sigma}{\rho_v}\right)^{1/2} \geq 0.079$ $\mathrm{Ku} = 14.1(d_e/l_e)\rho_v h_{lv}\left[C\left(\frac{\sigma}{\rho_v}\right)^{1/2}\right]^{1.15}$; $C\left(\frac{\sigma}{\rho_v}\right)^{1/2} < 0.079$ $C = 1.58\left(\frac{d_e}{\sigma}\right)^{0.4}$; $\frac{d_e}{\sigma} < 0.318$ $C = 1$; $\frac{d_e}{\sigma} \geq 0.318$
Gorbis and Savchenkov [89]	$\mathrm{Ku} = 0.0093(d_e/l_e)^{1.1}(d_e/l_e)^{-0.88}F_e^{-0.74} \times (1+0.03Bo)^2$; $2 < Bo < 60$
Bezrodnyi [90]	$\mathrm{Ku} = 2.55(d_e/l_e)\left\{\frac{\sigma}{p}\left[\frac{g(\rho_l-\rho_v)}{\sigma}\right]^{1/2}\right\}^{0.17}$; $\frac{\sigma}{p}\left[\frac{g(\rho_l-\rho_v)}{\sigma}\right]^{1/2} \geq 2.5\cdot10^{-5}$ $\mathrm{Ku} = 0.425(d_e/l_e)$; $\frac{\sigma}{p}\left[\frac{g(\rho_l-\rho_v)}{\sigma}\right]^{1/2} < 2.5\cdot10^{-5}$

D.A. Reay, P. Kew, Heat Pipes (5th ed.) 2006

Fig. 10: Closed two-phase TS: CCFL correlations (from [10])

Reference	Correlations Kutateladze number $\mathrm{Ku} = q_{\mathrm{ax}} / \boxed{\{\Delta h_{\mathrm{lv}} \rho_{\mathrm{v}}^{0.5} [\sigma g(\rho_{\mathrm{l}} - \rho_{\mathrm{v}})]^{0.25}\}}$
Groll and Rösler [91]	$\mathrm{Ku} = f_{1(Bo)} f_2 f_{3(\varphi, Bo)} (d_e/l_e)^{0.14}$ $f_2 = \left\{ \frac{\sigma}{p} \left[\frac{g(\rho_1 - \rho_v)}{\sigma} \right]^{1/2} \right\}^{0.17}; \quad \frac{\sigma}{p} \left[\frac{g(\rho_1 - \rho_v)}{\sigma} \right]^{1/2} \geq 2.5 \cdot 10^{-5}$ $f_2 = 0.165; \quad \frac{\sigma}{p} \left[\frac{g(\rho_1 - \rho_v)}{\sigma} \right]^{1/2} < 2.5 \cdot 10^{-5}$
Prenger [92]	$\mathrm{Ku} = 0.747 (d_e/l_e) [g\sigma(\rho_1 - \rho_v)]^{0.295} (h_{lv}\rho_v)^{-0.045}$
Fukano et al. [93]	$\mathrm{Ku} = 2 (d_e/l_e)^{0.83} F_e^{0.03} \left\{ \frac{[\sigma g(\rho_1 - \rho_v)]^{1/2}}{h_{lv}\rho_v} \right\}^{0.2}$
Imura et al. [94]	$\mathrm{Ku} = 0.16\{1 - \exp[-(d_e/l_e)(\rho_1/\rho_v)^{0.13}]\}$
Pioro and Voroncova [95]	$\mathrm{Ku} = 0.131\{1 - \exp[-(d_e/l_e)(\rho_1/\rho_v)^{0.13} \cos^{1.8}(\varphi - 55°)]\}^{0.8}$
Golobič and Gašperšič [96]	$q_{\mathrm{ax}} = \frac{0.16 d_e T_c^{1/3} p_c^{11/12} g^{1/4}}{l_e M^{1/4}} \tau \exp(2.530 - 8.233\tau + 1.387\omega + 17.096\tau^2 - 4.944\tau\omega^2 + 15.542\tau^2\omega^2 - 23.989\tau^3 - 19.235\tau^3\omega - 18.071\tau^3\omega^2$
D.A. Reay, P. Kew, Heat Pipes (5th ed.) 2006	$q_{\mathrm{ax}} = \frac{0.16 d_e T_c^{1/3} p_c^{11/12} g^{1/4}}{l_e M^{1/4}} \tau \exp(2.530 - 13.137\tau^2)$

Fig. 11: Closed two-phase TS: CCFL correlations (from [10]), continued

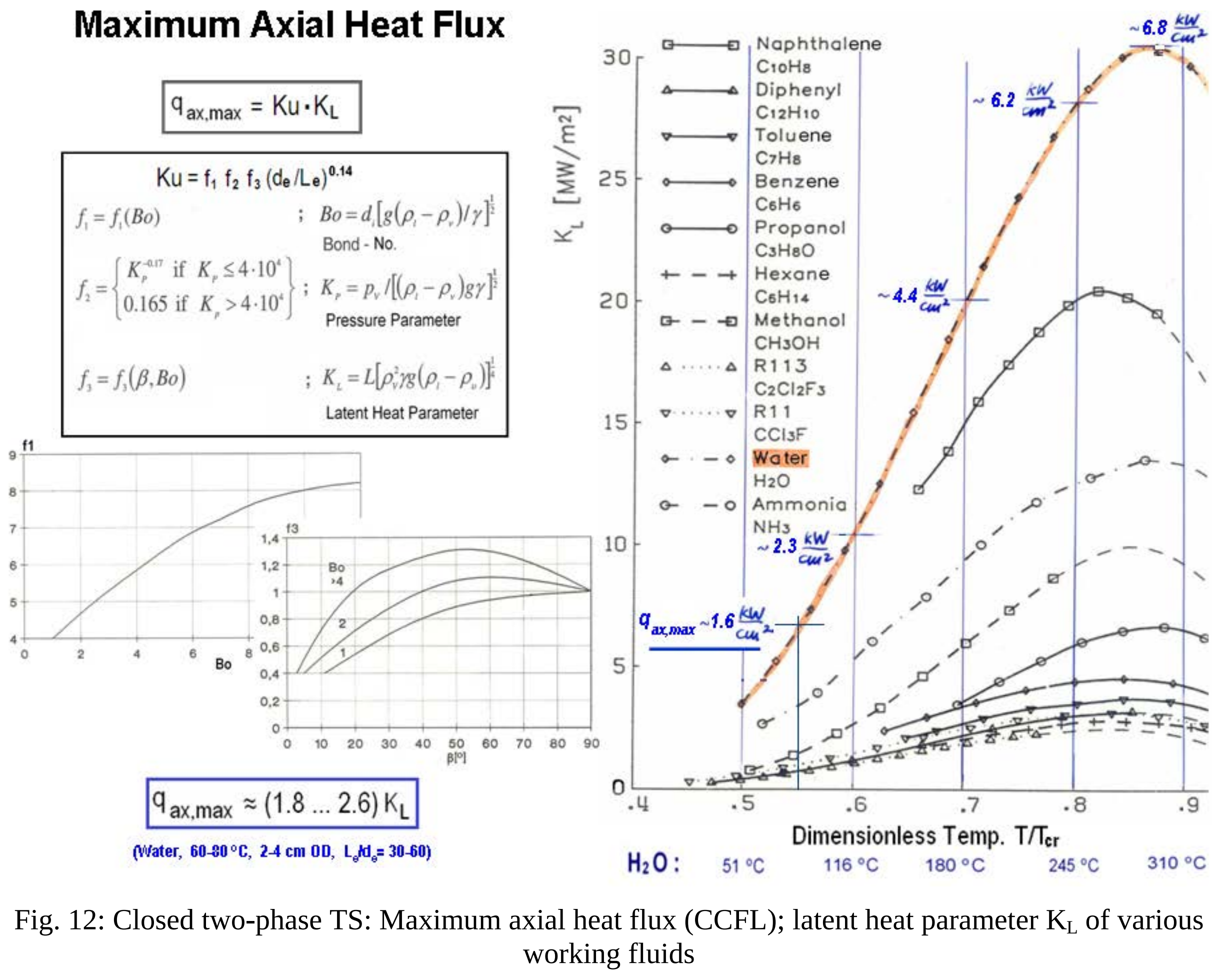


Fig. 12: Closed two-phase TS: Maximum axial heat flux (CCFL); latent heat parameter K_L of various working fluids

Four Generations of the Perkins Clan

(1) ***Jacob Perkins*** **(1766-1849): Single and two-phase heat transfer elements; various patents in 1836 on steam engines, furnaces, boilers, cooking devices, etc.; UK Pat. 7059 (1836): Perkins Tube (Closed Two-phase Thermosyphon)**

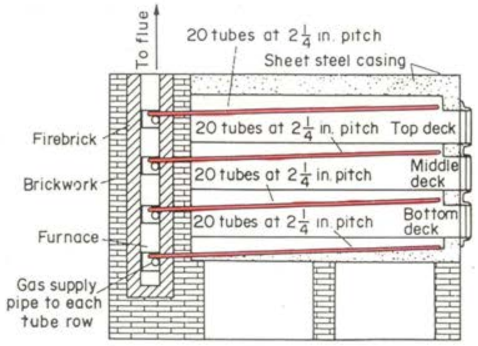
(2) ***Angier March Perkins*** **(1799-1881): Founder of A.M. Perkins & Son; Patents on Hermetic Tube Boiler (Single Phase Thermosyphon), UK Pat. 6146 (1831), UK Pat. 8311 (1839)**

(3a) *Angier Greenleaf Perkins* (1832-1871): engineer

(3b) *Loftus Perkins* (1834-1891): Partner in A.M. Perkins & Son

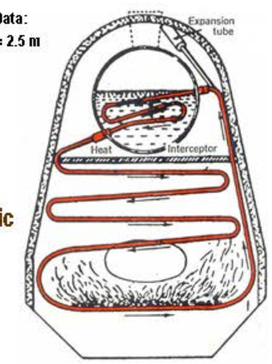
(4a) *Loftus Patton Perkins* (1868-1940)

(4b) ***Ludlow Patton Perkins*** **(1873-1928): Improvements and applications of the Perkins Tube; major new idea: Looped Perkins Tube. UK Patent 22272 (1892) (together with W.E. Buck)**



Baking Oven with Perkins Tubes

Typical Perkins Tube Data:
L_{tot} = 3 m, L_e = 5 cm, L_c = 2.5 m
D = 3cm, s = 5-6 mm
T = 210 – 230 C



Perkins Hermetic Tube Boiler
(Single Phase Thermosyphon)

Fig. 13: The Perkins clan (left), baking oven with Perkins Tubes (right, top), Perkins Hermetic Tube Boiler (right, bottom)

[Reprint.—*This Specification has been reprinted in consequence of an Amendment made by order of the Comptroller prior to the sealing of the Patent.*]

23 AUG.95

N° 22,272 A.D. 1892

Date of Application, 5th Dec., 1892
Complete Specification Left, 5th Oct., 1893—Accepted, 2nd Dec., 1893

PROVISIONAL SPECIFICATION.

Improvements in Devices for the Diffusion or Transference of Heat.

We, Ludlow Patton Perkins, of 43 Regent Square, in the Administrative County of London, and William Edward Buck, of Carisbrook, Battenhall, in the City of Worcester, in the County of Worcester, Engineers, do hereby declare the nature of this invention to be as follows :—

Our invention relates to improvements in devices for the diffusion or transference of heat.

It consists principally in a novel combination with lamps, stoves, furnaces, hearths, and other sources of heat, of a closed tube or tubes of suitable form or material and partially charged with liquid.

The liquid which we generally use is water, but we do not limit ourselves in this respect, but may make use of others which have the property of resisting the crystallizing influence of frost, or have boiling points higher or lower than that of water.

When it is desired to transmit heat of a low temperature and diffuse it evenly throughout the length of the tube or tubes, or to transmit heat of high temperature with great rapidity, we prefer to exhaust the tube or tubes of air as far as is practicable.

We construct the lamps, stoves or hearths, burning oil, gas, coke or other fuel, for use in combination with the above described closed heating tubes, in such manner that the latter can be easily removed for cleaning, packing, storing, or for regulating the heat transmitted, by withdrawing or partially withdrawing one or more tubes from the source of heat. The stoves are also so constructed and provided with stops and guides that, if the tubes are removed, they can only be replaced at angles proper to their working and with suitable portions of them exposed to the action of the heat.

The tubes as above described are preferably sealed at both ends; but, for special purposes, where it is desired to transmit heat to considerable distances in small tubes, we bend and return the tube into itself in such manner that the outgoing current does not meet the return flow.

The above invention may, if suitably modified, be advantageously applied to very many purposes of which the following are some.

For heating greenhouses, rooms, vehicles, drying closets and the like, its use ensures the rapid production, transmission and diffusion of heat, together with long continuance, ease and rapidity of regulation, and rapid loss of the accumulated heat when its source is removed.

The lamps or stoves used for the above purposes may be constructed so that the products of combustion are carried outside the house, chamber or vehicle, or are condensed as far as possible, in the lamp or stove itself.

Those portions of the tubes which are outside the lamp or stove may either be exposed or may be surrounded by an ornamental casing or trellis.

[Price 6d.]

Patent of L.P. Perkins & W.E. Buck

Improvements

Straight , slightly elevated tubes
Loop thermosyphon to separate phases

Applications

- Indirect heating of boilers for steam engines, locomotives, ...
- Heating of greenhouses, rooms , windows, currents or volumes of air or liquids dairy products, chemicals, ...
- Cooling of liquids to ambient temperature of cooling water for many machines, ...
- Waste heat recovery (from exhaust gas of blast furnace to preheat incoming air)

Fig. 14: Patent of L.P. Perkins and W.E. Buck: first page

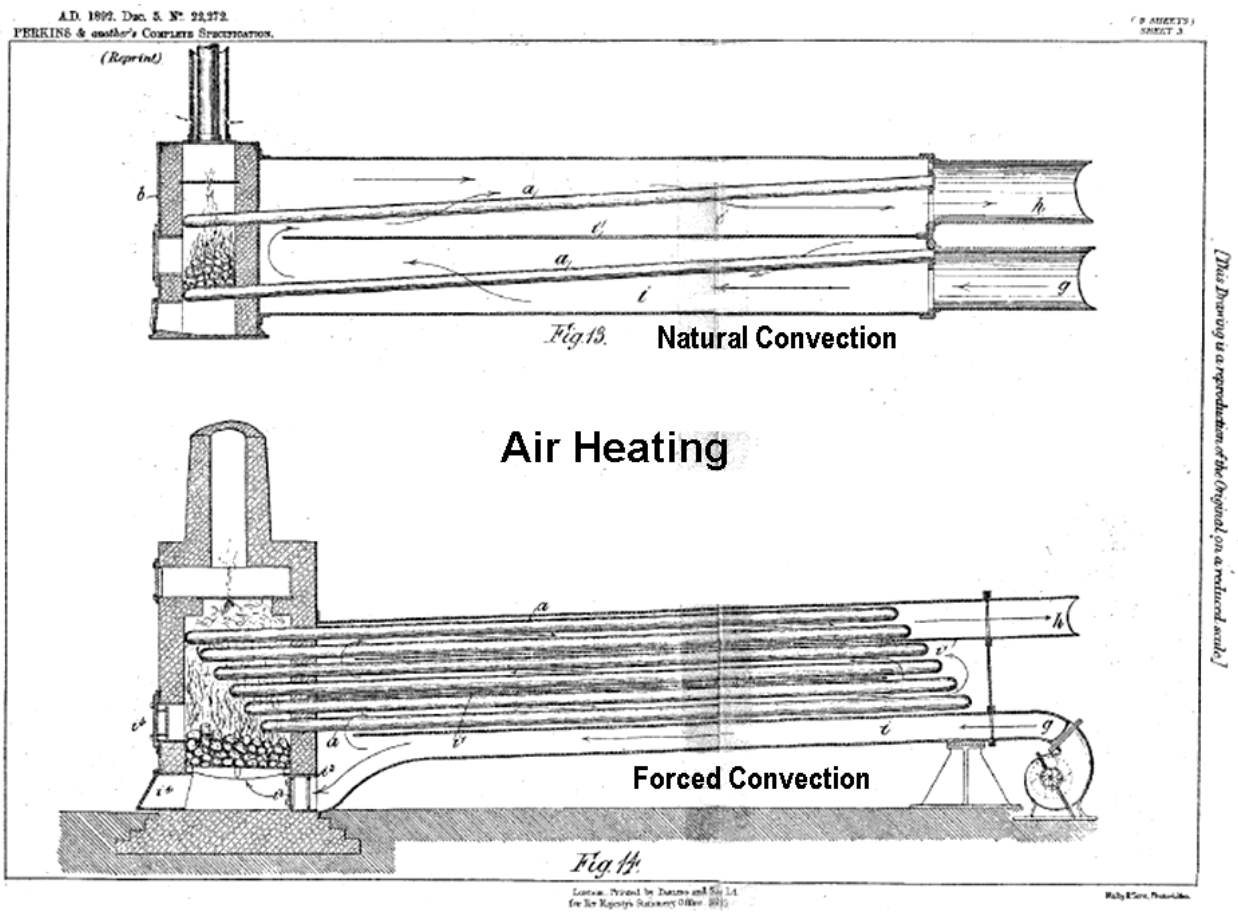


Fig. 15: Patent of L.P. Perkins and W.E. Buck: air heating with bundle of Perkins Tubes

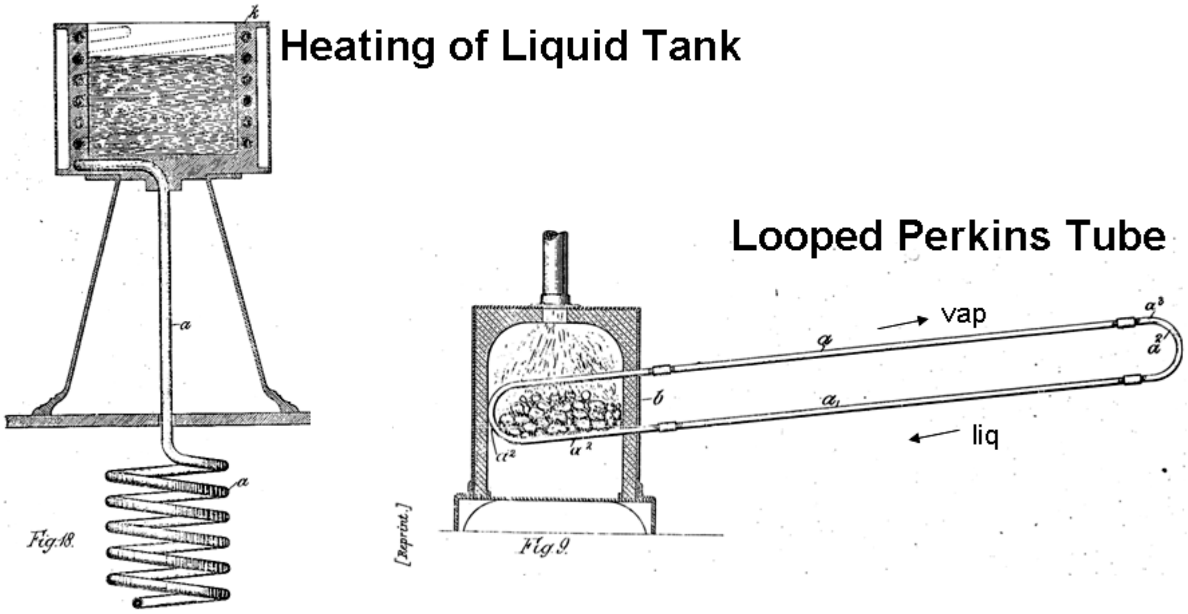


Fig. 16: Patent of L.P. Perkins and W.E. Buck: heating of liquid tank with Perkins Tube (left); Looped Perkins Tube (right)

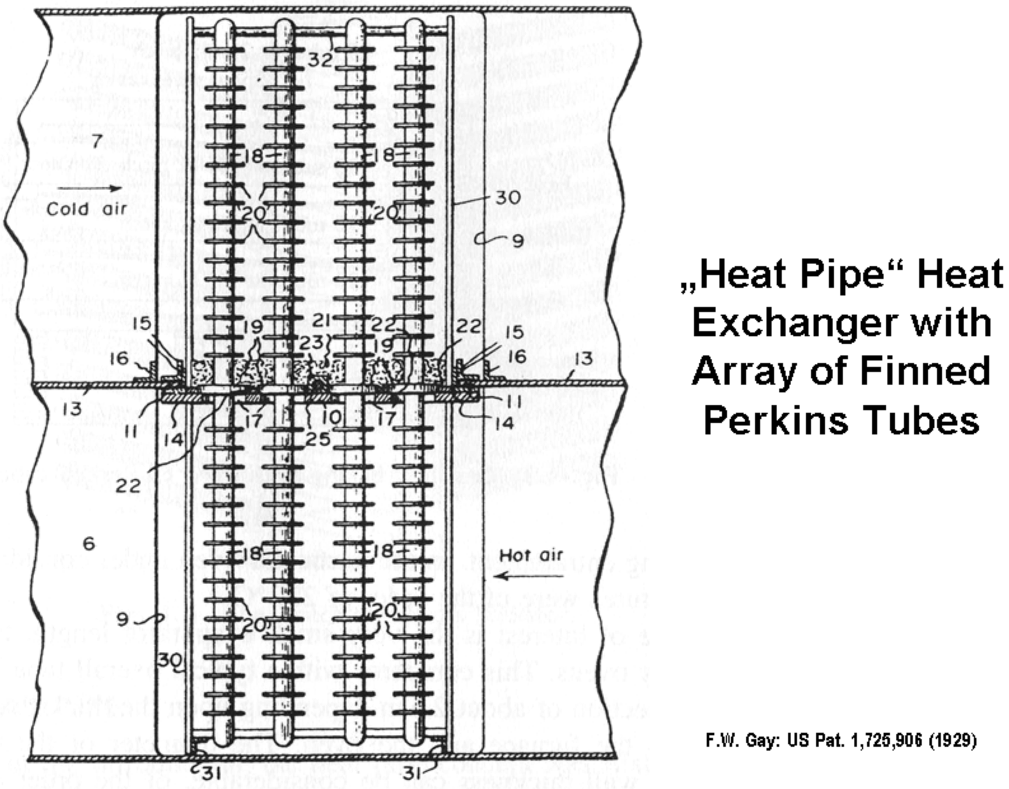


Fig. 17: Patent of F.W. Gray: “Heat pipe” heat exchanger with finned Perkins Tubes

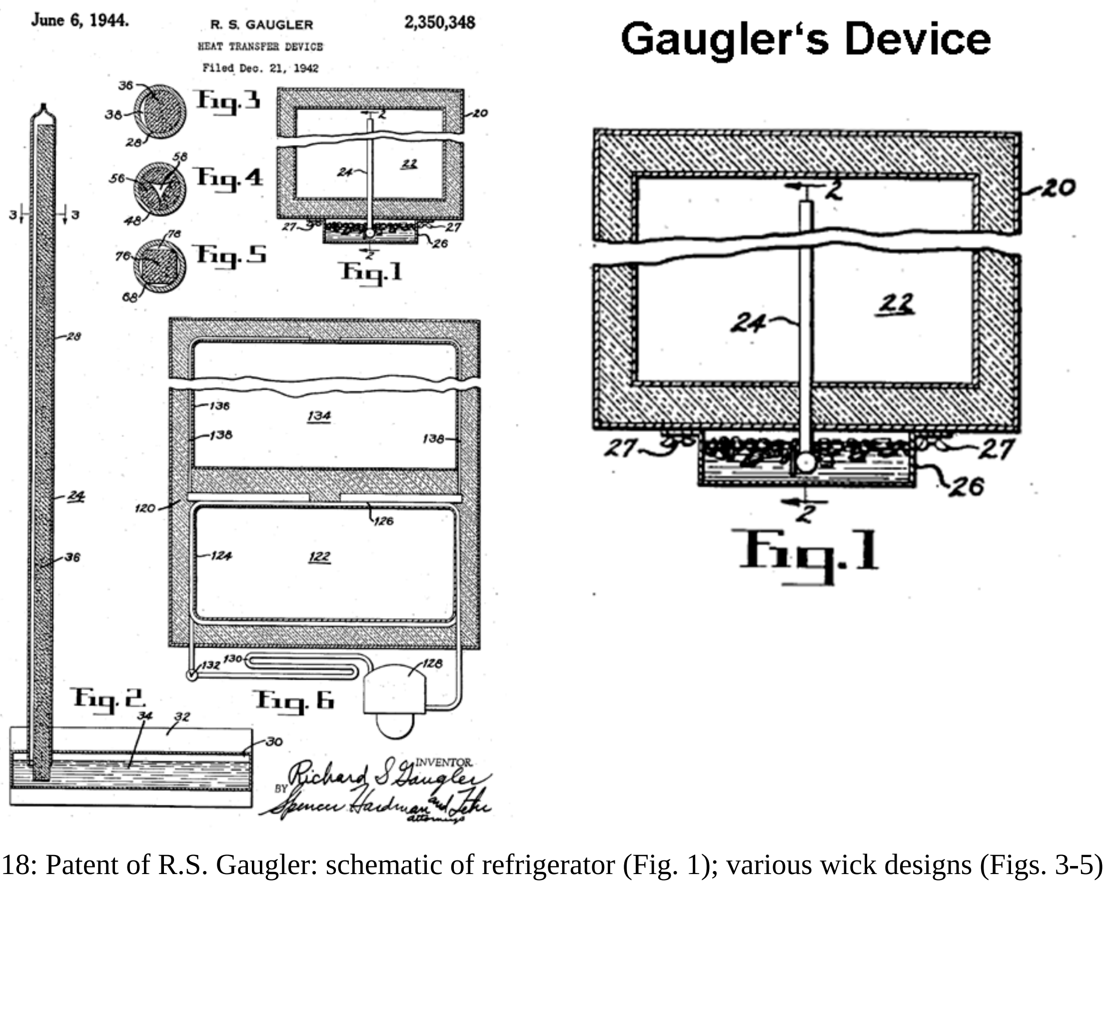
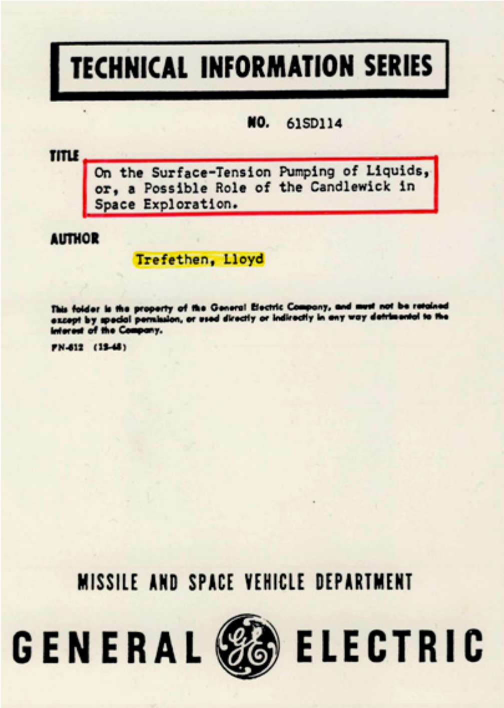


Fig. 18: Patent of R.S. Gaugler: schematic of refrigerator (Fig. 1); various wick designs (Figs. 3-5)


TECHNICAL INFORMATION SERIES

NO. 61SD114

TITLE

On the Surface-Tension Pumping of Liquids, or, a Possible Role of the Candlewick in Space Exploration.

AUTHOR

Trefethen, Lloyd

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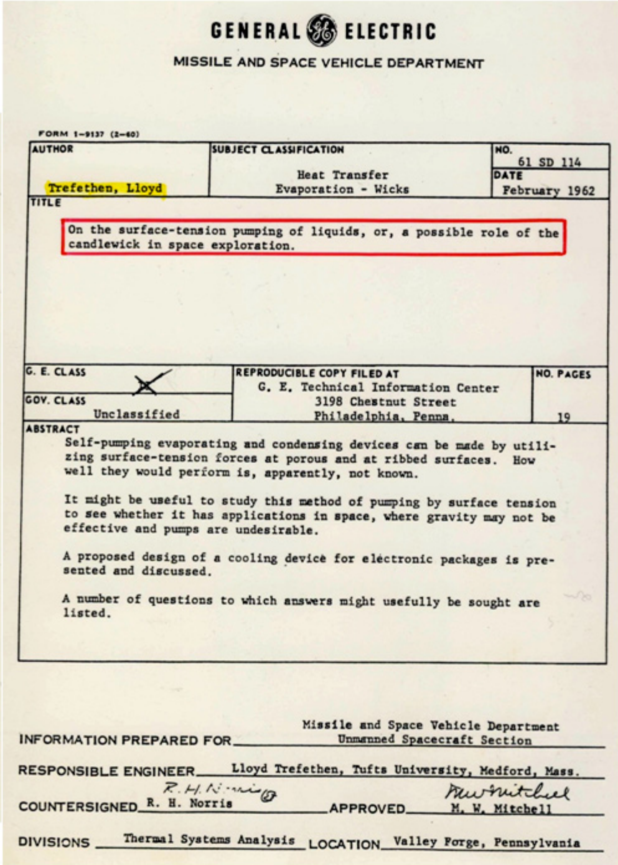
FN-612 (12-48)

MISSILE AND SPACE VEHICLE DEPARTMENT

GENERAL ELECTRIC


GENERAL ELECTRIC

MISSILE AND SPACE VEHICLE DEPARTMENT

FORM 1-9137 (2-60)

AUTHOR	SUBJECT CLASSIFICATION	NO. / DATE
Trefethen, Lloyd	Heat Transfer Evaporation - Wicks	61 SD 114 / February 1962

TITLE

On the surface-tension pumping of liquids, or, a possible role of the candlewick in space exploration.

G. E. CLASS	REPRODUCIBLE COPY FILED AT	NO. PAGES
X	G. E. Technical Information Center 3198 Chestnut Street Philadelphia, Penna.	19
GOV. CLASS Unclassified		

ABSTRACT

Self-pumping evaporating and condensing devices can be made by utilizing surface-tension forces at porous and at ribbed surfaces. How well they would perform is, apparently, not known.

It might be useful to study this method of pumping by surface tension to see whether it has applications in space, where gravity may not be effective and pumps are undesirable.

A proposed design of a cooling device for electronic packages is presented and discussed.

A number of questions to which answers might usefully be sought are listed.

INFORMATION PREPARED FOR Missile and Space Vehicle Department Unmanned Spacecraft Section

RESPONSIBLE ENGINEER Lloyd Trefethen, Tufts University, Medford, Mass.

COUNTERSIGNED R. H. Norris APPROVED M. W. Mitchell

DIVISIONS Thermal Systems Analysis LOCATION Valley Forge, Pennsylvania

Fig. 19: Report of L. Trefethen: cover page and abstract

Grover's Lab Notebook (July 24, 1963) p.101

Heat transfer via capillary movement of fluids.

The "pumping" action of surface tension forces may be sufficient to move liquids from a cold temperature zone to a high temperature zone (with subsequent return in vapor form using as the driving force, the difference in vapor pressure at the two temperatures) to be of interest in transferring heat from the hot to the cold zone. Such a closed system, requiring no external pumps may be of particular interest in heat from the reactor core to a radiating system. In the absence of gravity, the forces must only be such as to overcome the viscous drag of the fluid through the capillary and the drag of the returning vapor through its channels.

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Fig. 20: Lab notebook of G.M. Grover: page on heat pipe principle (left); Grover testing a Na heat pipe (right bottom)

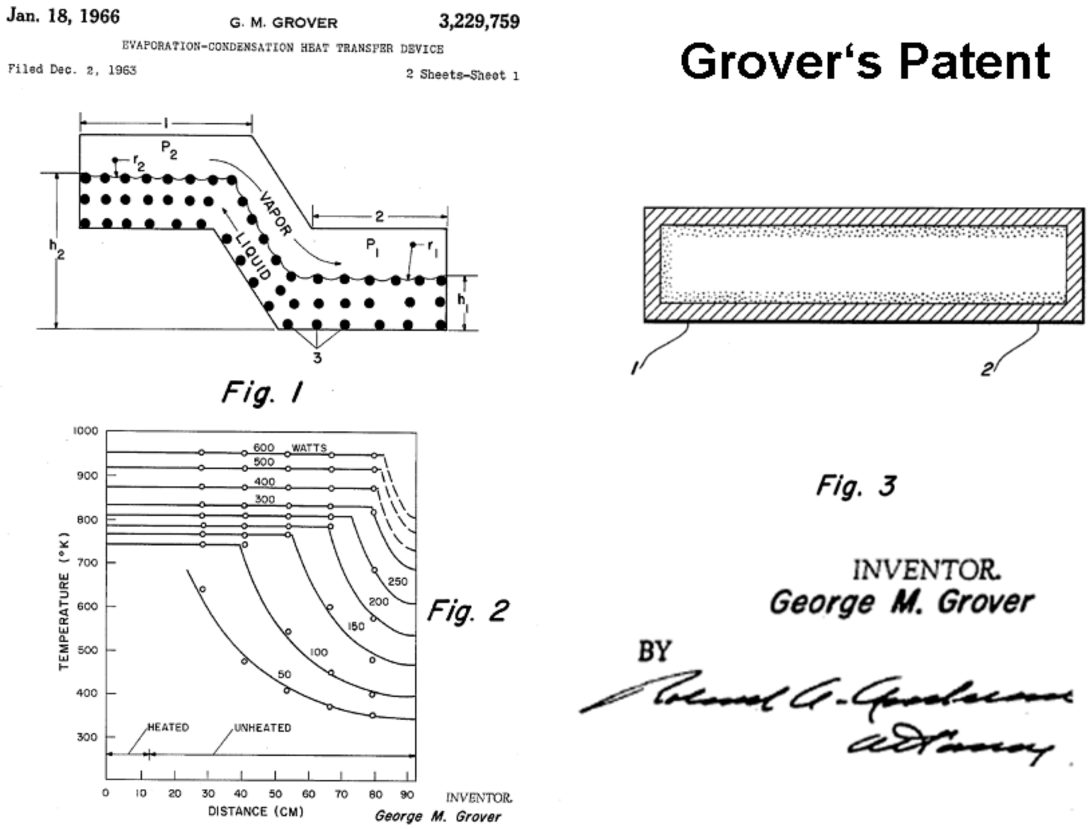
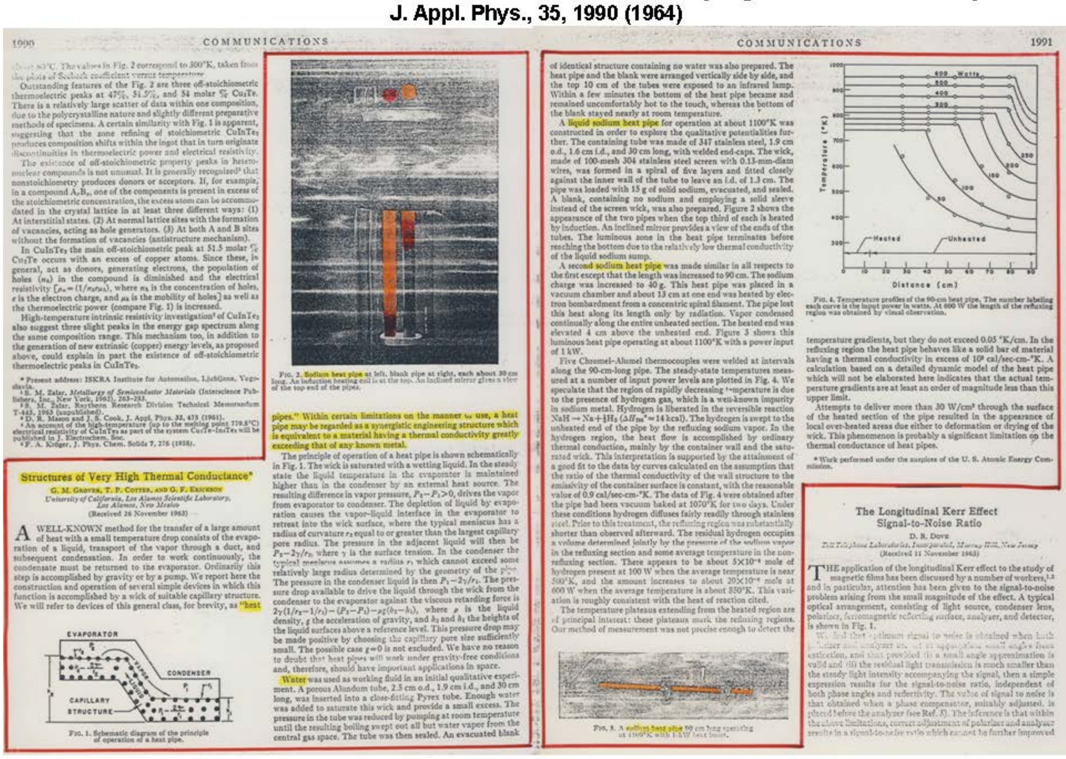


Fig. 21: Patent of G.M. Grover: operation principle of heat pipe (Fig. 1); experimental results of Na heat pipe (Fig. 2)

Grover, G. M., Cotter, T. P. and Erikson, G. F., "Structures of Very High Thermal Conductivity", J. Appl. Phys., 35, 1990 (1964)



1990 COMMUNICATIONS

Structures of Very High Thermal Conductance*

G. M. Grover, T. P. Cotter, and G. F. Erickson
University of California, Los Alamos Scientific Laboratory, Los Alamos, New Mexico
(Received 26 November 1963)

A WELL-KNOWN method for the transfer of a large amount of heat with a small temperature drop consists of the evaporation of a liquid, transport of the vapor through a duct, and subsequent condensation. In order to work continuously, the condensate must be returned to the evaporator. Ordinarily this step is accomplished by gravity or by a pump. We report here the construction and operation of several simple devices in which this function is accomplished by a wick of suitable capillary structure. We will refer to devices of this general class, for brevity, as "heat pipes." Within certain limitations on the manner of use, a heat pipe may be regarded as a synergistic engineering structure which is equivalent to a material having a thermal conductivity greatly exceeding that of any known metal.

The principle of operation of a heat pipe is shown schematically in Fig. 1. The wick is saturated with a wetting liquid. In the steady state the liquid temperature in the evaporator is maintained higher than in the condenser by an external heat source. The resulting difference in vapor pressure, $P_2-P_1>0$, drives the vapor from evaporator to condenser. The depletion of liquid by evaporation causes the vapor-liquid interface in the evaporator to retreat into the wick surface, where the typical meniscus has a radius of curvature r_2 equal to or greater than the largest capillary pore radius. The pressure in the adjacent liquid will then be $P_2-2\gamma/r_2$, where γ is the surface tension. In the condenser the typical meniscus assumes a radius r_1 which cannot exceed some relatively large radius determined by the geometry of the pipe. The pressure in the condenser liquid is then $P_1-2\gamma/r_1$. The pressure drop available to drive the liquid through the wick from the condenser to the evaporator against the viscous retarding force is $2\gamma(1/r_2-1/r_1)-(P_2-P_1)-\rho g(h_2-h_1)$, where ρ is the liquid density, g the acceleration of gravity, and h_2 and h_1 the heights of the liquid surfaces above a reference level. This pressure drop may be made positive by choosing the capillary pore size sufficiently small. The possible case $g=0$ is not excluded. We have no reason to doubt that heat pipes will work under gravity-free conditions and, therefore, should have important applications in space.

Water was used as working fluid in an initial qualitative experiment. A porous Alundum tube, 2.5 cm o.d., 1.9 cm i.d., and 30 cm long, was inserted into a close-fitting Pyrex tube. Enough water was added to saturate this wick and provide a small excess. The pressure in the tube was reduced by pumping at room temperature until the resulting boiling swept out all but water vapor from the central gas space. The tube was then sealed. An evacuated blank of identical structure containing no water was also prepared. The heat pipe and the blank were arranged vertically side by side, and the top 10 cm of the tubes were exposed to an infrared lamp. Within a few minutes the bottom of the heat pipe became and remained uncomfortably hot to the touch, whereas the bottom of the blank stayed nearly at room temperature.

A liquid sodium heat pipe for operation at about 1100°K was constructed in order to explore the qualitative potentialities further. The containing tube was made of 347 stainless steel, 1.9 cm o.d., 1.6 cm i.d., and 30 cm long, with welded end-caps. The wick, made of 100-mesh 304 stainless steel screen with 0.13-mm-diam wires, was formed in a spiral of five layers and fitted closely against the inner wall of the tube to leave an i.d. of 1.3 cm. The pipe was loaded with 15 g of solid sodium, evacuated, and sealed. A blank, containing no sodium and employing a solid sleeve instead of the screen wick, was also prepared. Figure 2 shows the appearance of the two pipes when the top third of each is heated by induction. An inclined mirror provides a view of the ends of the tubes. The luminous zone in the heat pipe terminates before reaching the bottom due to the relatively low thermal conductivity of the liquid sodium sump.

A second sodium heat pipe was made similar in all respects to the first except that the length was increased to 90 cm. The sodium charge was increased to 40 g. This heat pipe was placed in a vacuum chamber and about 13 cm at one end was heated by electron bombardment from a concentric spiral filament. The pipe lost this heat along its length only by radiation. Vapor condensed continually along the entire unheated section. The heated end was elevated 4 cm above the unheated end. Figure 3 shows this luminous heat pipe operating at about 1100°K with a power input of 1 kW.

Five Chromel–Alumel thermocouples were welded at intervals along the 90-cm-long pipe. The steady-state temperatures measured at a number of input power levels are plotted in Fig. 4. We speculate that the region of rapidly decreasing temperature is due to the presence of hydrogen gas, which is a well-known impurity in sodium metal. Hydrogen is liberated in the reversible reaction $NaH \rightarrow Na+\frac{1}{2}H_2$ ($\Delta H_{298}° \approx 14$ kcal). The hydrogen is swept to the unheated end of the pipe by the refluxing sodium vapor. In the hydrogen region, the heat flow is accomplished by ordinary thermal conduction, mainly by the container wall and the saturated wick. This interpretation is supported by the attainment of a good fit to the data by curves calculated on the assumption that the ratio of the thermal conductivity of the wall structure to the emissivity of the container surface is constant, with the reasonable value of 0.9 cal/sec-cm-°K. The data of Fig. 4 were obtained after the pipe had been vacuum baked at 1070°K for two days. Under these conditions hydrogen diffuses fairly readily through stainless steel. Prior to this treatment, the refluxing region was substantially shorter than observed afterward. The residual hydrogen occupies a volume determined jointly by the pressure of the sodium vapor in the refluxing section and some average temperature in the nonrefluxing section. There appears to be about 5×10^{-6} mole of hydrogen present at 100 W when the average temperature is near 500°K, and the amount increases to about 20×10^{-6} mole at 600 W when the average temperature is about 850°K. This variation is roughly consistent with the heat of reaction cited.

The temperature plateaus extending from the heated region are of principal interest: these plateaus mark the refluxing regions. Our method of measurement was not precise enough to detect the temperature gradients, but they do not exceed 0.05 °K/cm. In the refluxing region the heat pipe behaves like a solid bar of material having a thermal conductivity in excess of 10^3 cal/sec-cm-°K. A calculation based on a detailed dynamic model of the heat pipe which will not be elaborated here indicates that the actual temperature gradients are at least an order of magnitude less than this upper limit.

Attempts to deliver more than 30 W/cm² through the surface of the heated section of the pipe resulted in the appearance of local over-heated areas due either to deformation or drying of the wick. This phenomenon is probably a significant limitation on the thermal conductance of heat pipes.

* Work performed under the auspices of the U. S. Atomic Energy Commission.

Fig. 1. Schematic diagram of the principle of operation of a heat pipe.

Fig. 2. Sodium heat pipe at left, blank pipe at right, each about 30 cm long. An induction heating coil is at the top. An inclined mirror gives a view of the top end of the pipes.

Fig. 3. A sodium heat pipe 90 cm long operating at 1100°K with 1-kW heat input.

Fig. 4. Temperature profiles of the 90-cm heat pipe. The number labeling each curve is the input power in watts. At 600 W the length of the refluxing region was obtained by visual observation.

COMMUNICATIONS 1991

The Longitudinal Kerr Effect Signal-to-Noise Ratio

D. B. Dove
(Received 11 November 1963)

THE application of the longitudinal Kerr effect to the study of magnetic films has been discussed by a number of workers,[1,2] and in particular, attention has been given to the signal-to-noise problems arising from the small magnitude of the effect. A typical

Fig. 22: First paper on heat pipes by G.M. Grover, T.P. Cotter and G.F. Ericson

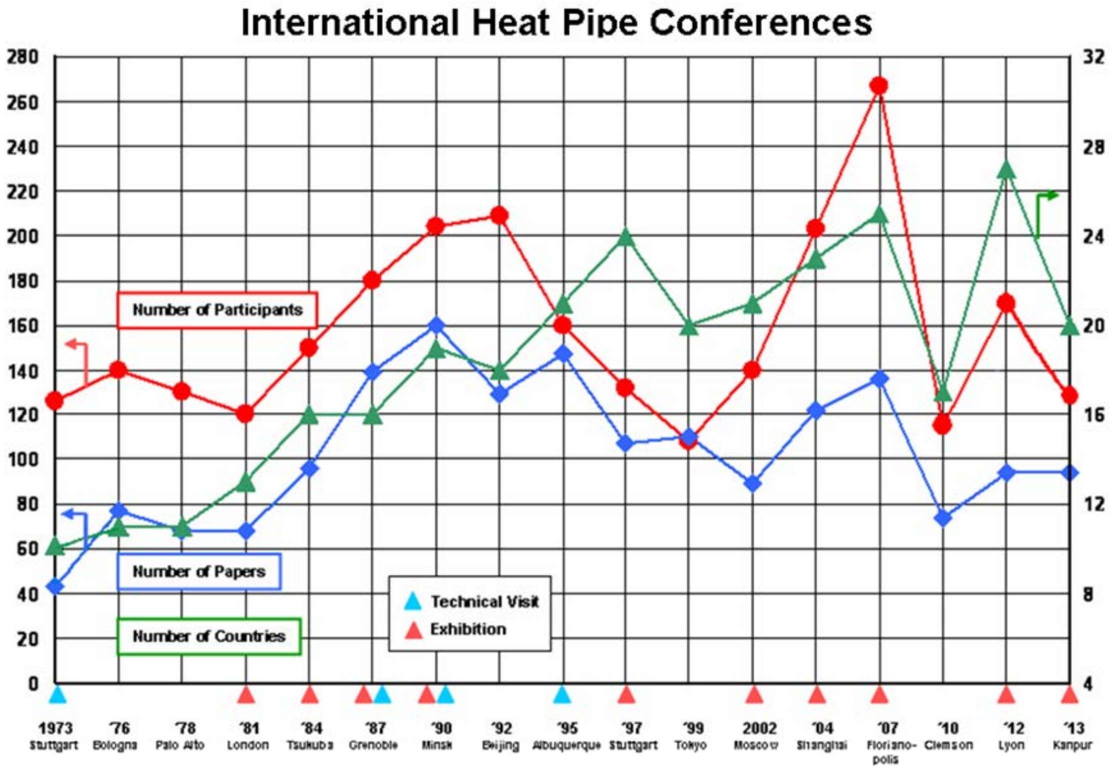


Fig. 23: Statistics of IHPC

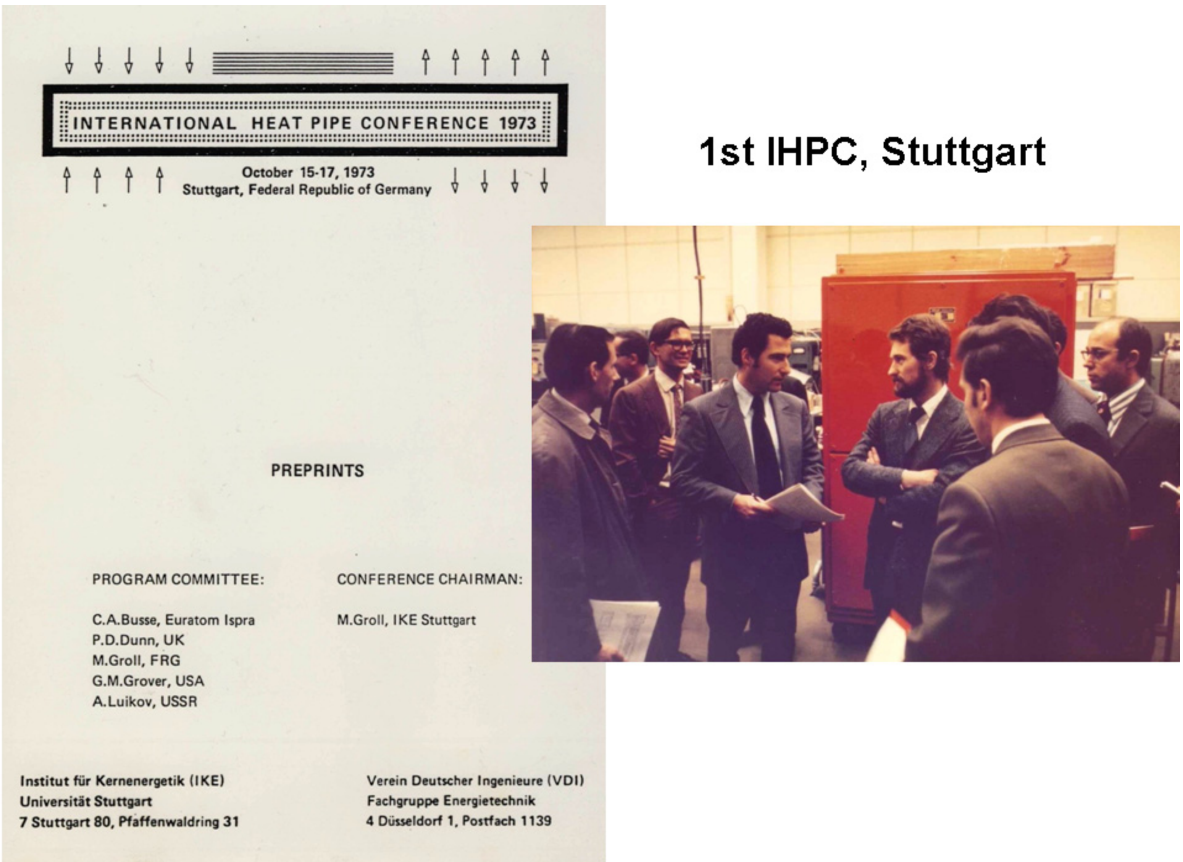


Fig. 24: 1st IHPC in Stuttgart, 1973: cover page of preprints (left), photo of IKE lab visit (right)

George M. Grover
at 2nd IHPC, Bologna

Fig. 25: G.M. Grover at 2nd IHPC in Bologna, 1976

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Cotter, T.P., *Theory of Heat Pipes*, Los Alamos Scientific Laboratory, Rept. LA-3246-MS, 1965

Marcus, B.D., *On the Operation of Heat Pipes*, TRW Rept. 9895-6001-TU-000, 1965

Cheung, H., *A Critical Review of Heat Pipe Theory* and *Applications*, Lawrence Radiation Laboratory, TID-4500, UC-4 (Chemistry), 1968

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Brennan, P.J., Kroliczek, E.J., *Heat Pipe Design Handbook*, NASA Contract No. NAS5-23406, 1979

ESDU, *Heat Pipes - Performance of Closed Two-Phase Thermosyphons*, ESDU Item No. 81038 Engineering Sciences Data Unit, London, 1981

Ivanovskii, M.N., Sorokin, V.P., Yagadkin, I.V., *The Physical Principles of Heat Pipes*, Clarendon Press, Oxford, 1982

Terpstra, M., Van Veen, J.G., *Heat Pipes: Construction and Application*, Elsevier Appl. Sci., London, 1987

Groll, M., Roesler, S., *Operation Principles of Heat Pipes and Closed Two-Phase Thermosyphons*, in: J. Non-Equilib. Thermodyn., Vol. 17, p.91-151,1992

Silverstein, C., *Design and Technology of Heat Pipes for Cooling and Heat Exchange*, Taylor & Francis, Washington, D.C., 1992

Peterson, G.P., *An Introduction to Heat Pipes (Modeling, Testing and Applications)*, Wiley & Sons, Inc., Hoboken, N.J., 1994

Faghri, A., *Heat Pipe Science and Technology*, Taylor & Francis, Washington, D.C., 1995

Pioro, L.S., Pioro, I.L., *Industrial Two-Phase Thermosyphons*, Begell House, Inc., 1997

Groll, M, Khandekar, S., *Heat Transfer in Microchannels: Micro Heat Pipes*, in: Heat Exchanger Design Handbook (Update),Chapter on Microscale Boiling & Condensation, Begell House, Vol.9, Issue 1&2, 2002

Ochterbeck, J., *Heat Pipes*, in: Heat Transfer Handbook,Chapter 16, p.1181-1231, Wiley & Sons, Inc., Hoboken, N.J., 2003

Fig. 26: Review reports and textbooks on heat pipes

Major Events in Heat Pipe Science & Development

J.Perkins: Single and Two-phase Thermosyphons, UK Patents in 1836, Perkins Tube
L.P.Perkins, W.E.Buck: Improvements of Perkins Tube, UK Patent, 1892
Gay: Heat Exchanger with Finned Perkins Tubes, US Patent, 1929
Anderson: Anti-gravity Thermosyphon, US Patent, 1940
Gaugler: Anti-gravity Heat Pipe, US Patent, 1942
Trefethen: Zero-gravity Heat Pipe, G.E. Internal Report, 1962
Grover: Zero-gravity Heat Pipe, US Patent, 1963
Grover et al.: Zero-gravity Heat Pipe, Publication 1964
Cotter: Theory of Heat Pipes, LASL Report, 1965
Deverall, Kemme: Satellite Tests of Heat Pipes, 1967
Haskin: Cryogenic Heat Pipe, 1967
Gray: Rotating Heat Pipe, US Patent, 1969
Turner, Bienert: Heat Pipes for Temperature Control, 1969
Marcus: Theory and Design of Variable Concuctance Heat Pipes, 1972

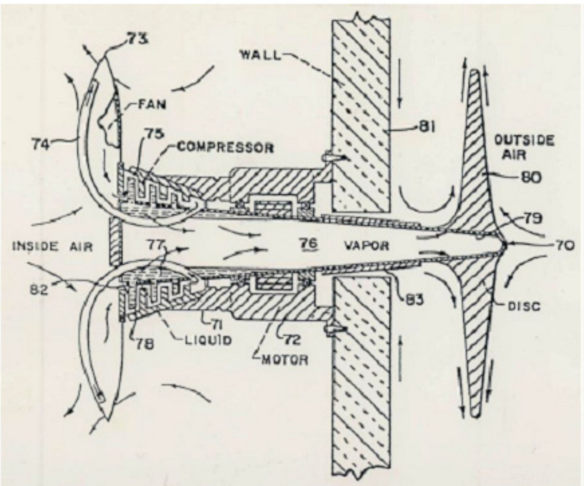
NASA ARC: Temperature Control with Heat Pipes, Space Experiments, early 1970's
- Ames Heat Pipe Experiment (AHPE) on OAO-3, 1972
- Advanced Thermal Control Flight Experiment (ATFE) on ATS-F, 1974

NASA ARC: Innovative Heat Pipe Designs, 1970's
- Electrohydrodynamic Heat Pipe
- Artery Heat Pipes and Priming Devices
- Inverted Meniscus Heat Pipe

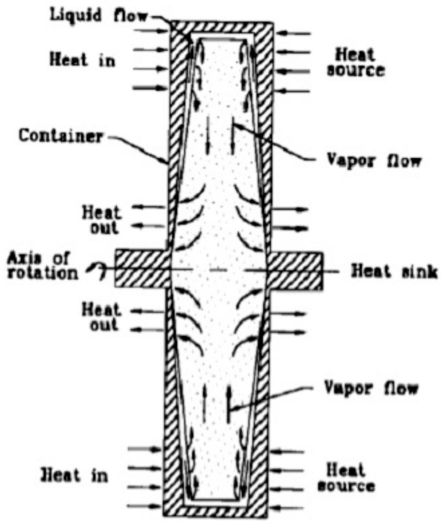
USA: Alaska Oil Pipeline, mid 1970's
UK, USA : Anti-gravity Thermosyphons, mid 1970's – 80's
Stenger : Capillary Pumped Loops, 1970's/80's, 1st Flight Experiments, 1985, 1986
Gerassimov, Maydanik: Loop Heat Pipes, early 1980's, 1st Flight Experiment, 1989
Japan, PRC: Separate-type Heat Pipes, mid 1980's
Cotter: Micro Heat Pipe, 1984
Akachi: Oscillating (Pulsating) Heat Pipe, US Patents, 1990 - 1996

Fig. 27: Major events in heat pipe science and technology

Rotating Heat Pipes



Rotating Heat Pipe: Hollow Shaft



Rotating Heat Pipe: Hollow Disc

Fig. 28: Schematics of rotating heat pipes

Heat Pipes for Temperature Control

Variable Conductance Heat Pipes

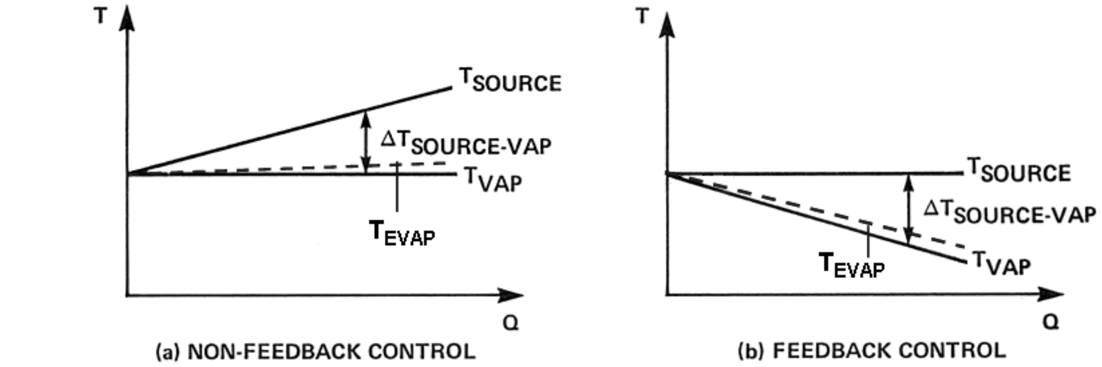


Fig. 29: Principle of heat pipe temperature control

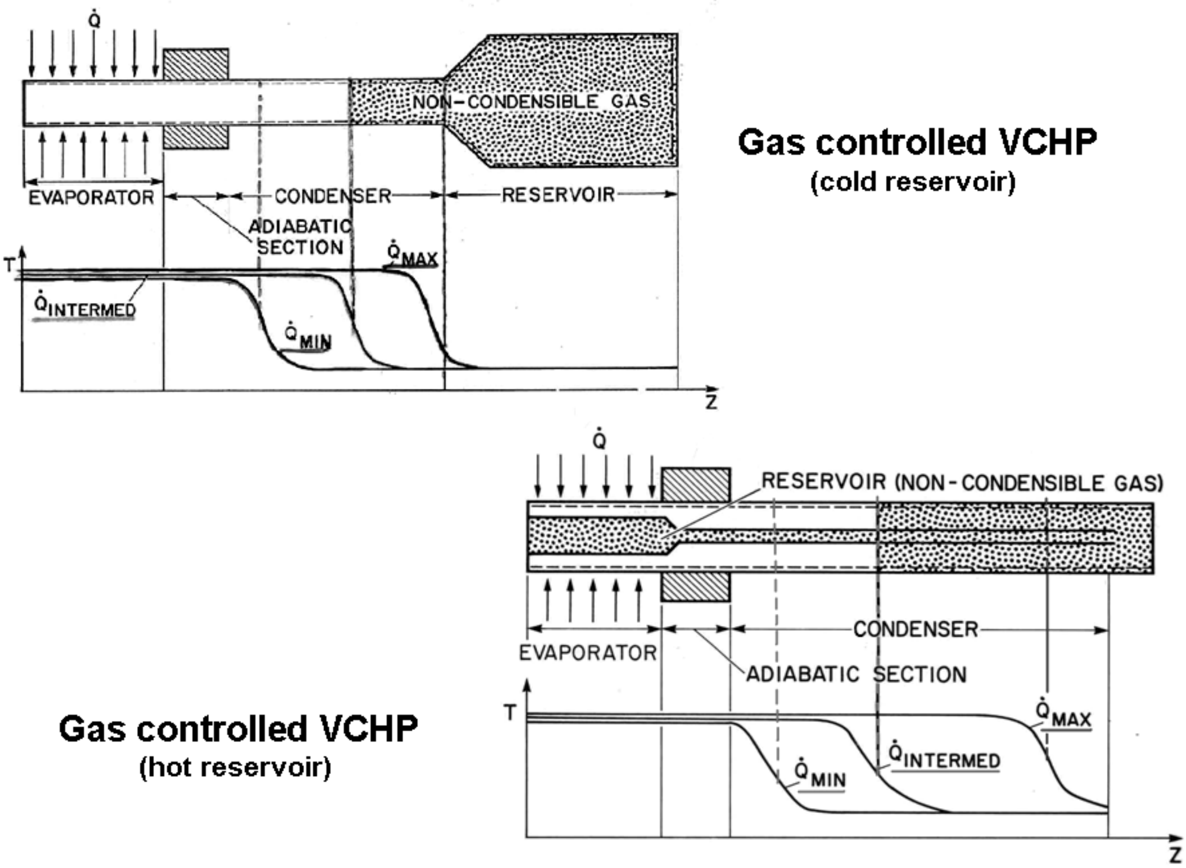


Fig. 30: Non-feedback gas controlled VCHP: schematic and operation principle

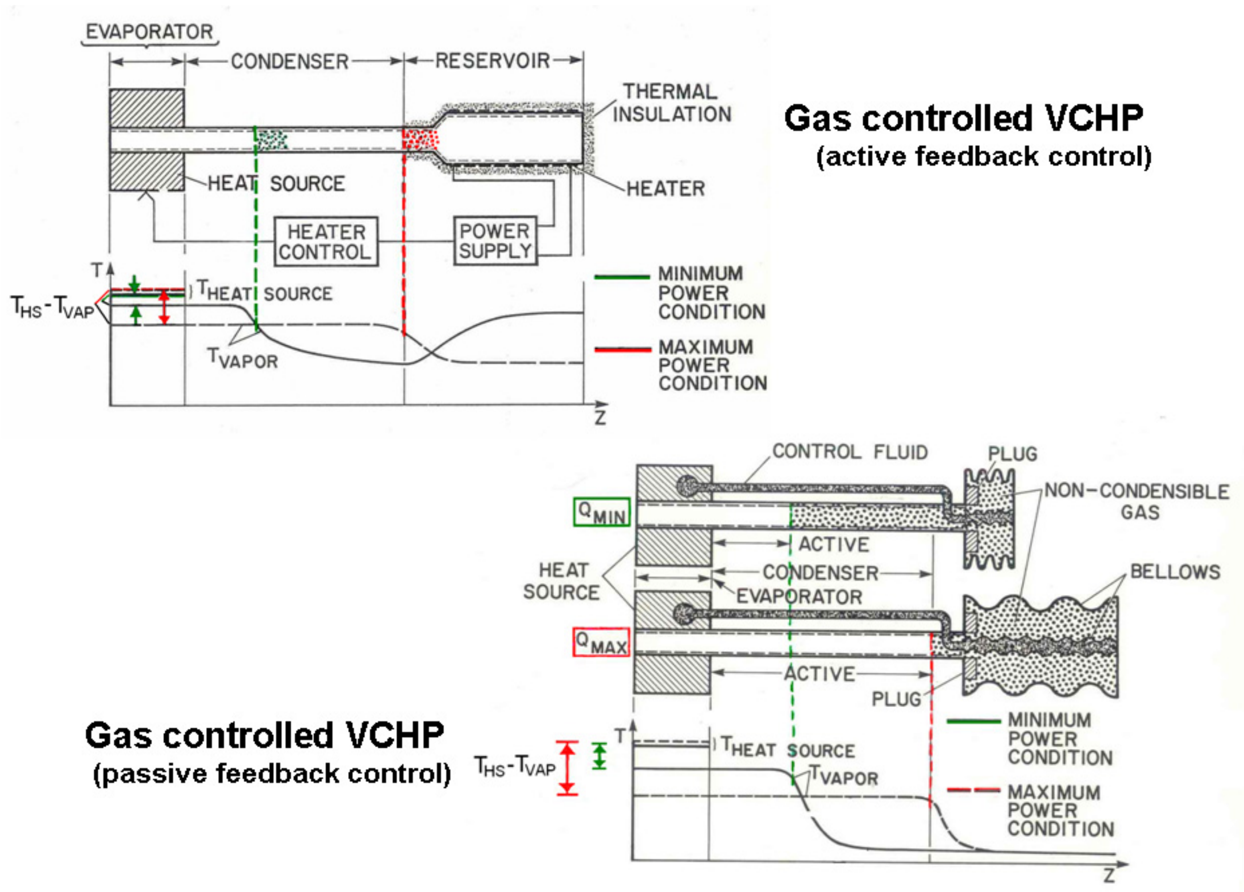


Fig. 31: Feedback gas controlled VCHP: schematic and operation principle

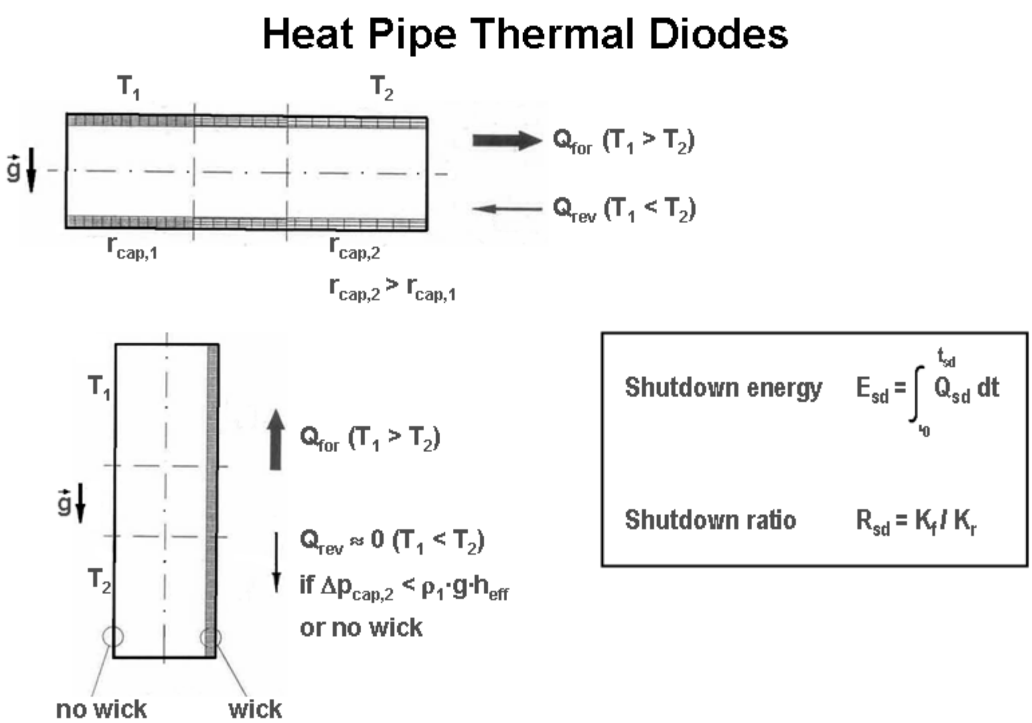


Fig. 32: Heat pipe thermal diodes: capillarity control (top) and gravity control (bottom)

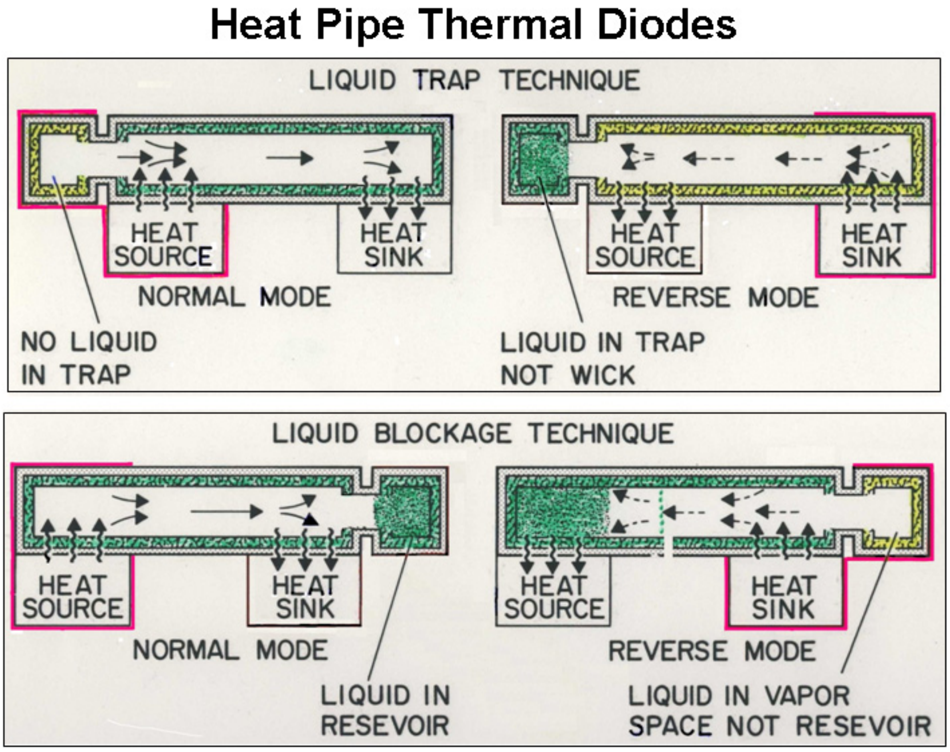


Fig. 33: Heat pipe thermal diodes: liquid trap and liquid blockage technique

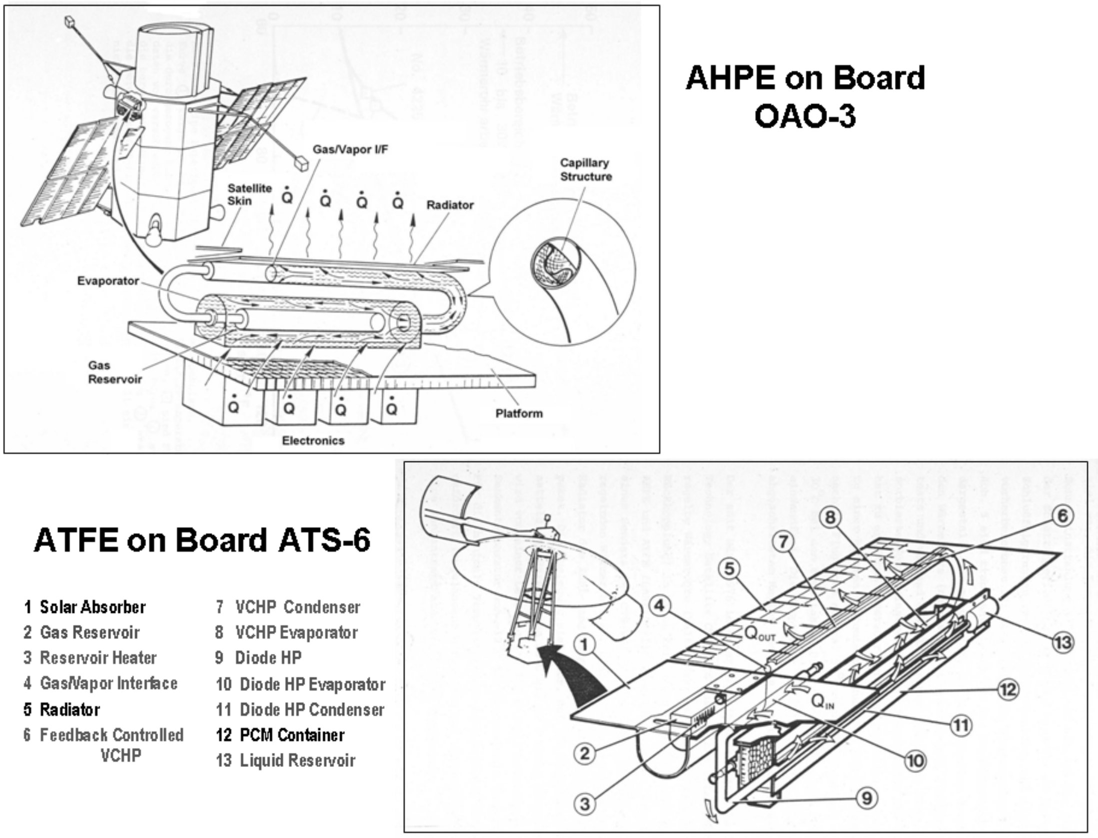


Fig. 34: Temperature control experiments in space: AHPE (left), ATFE (right)

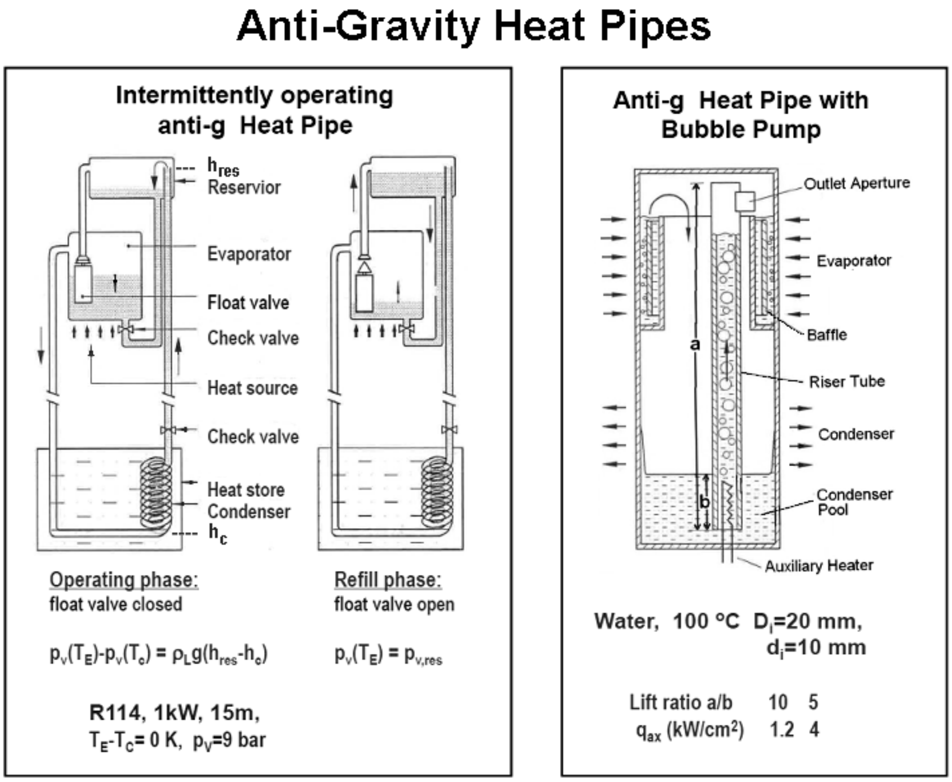


Fig. 35: Anti-g heat pipes: intermittent operation (left), vapour lift pump (right)

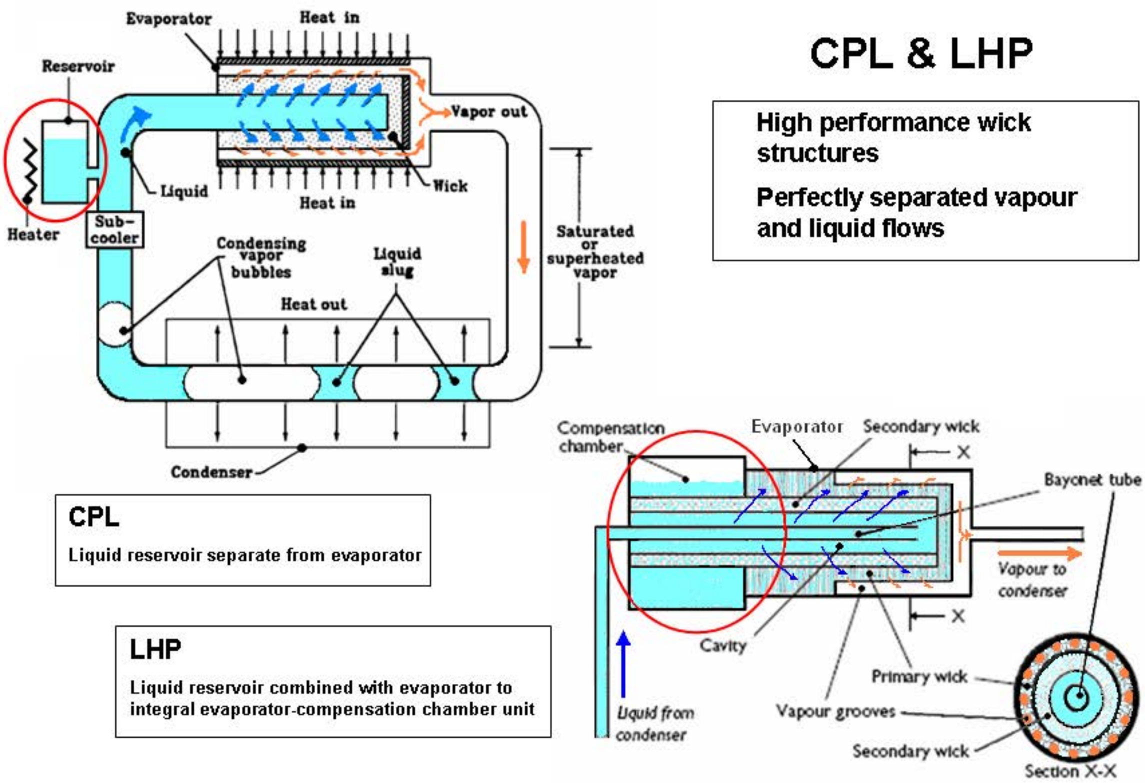
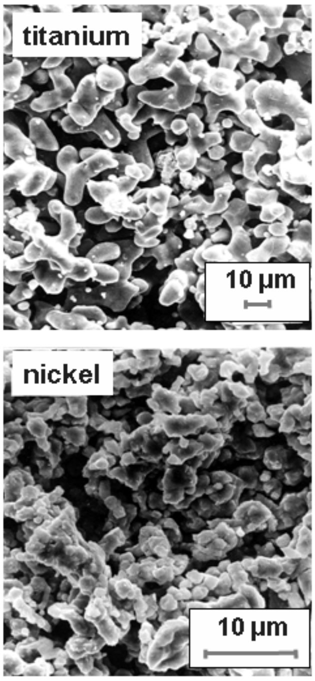
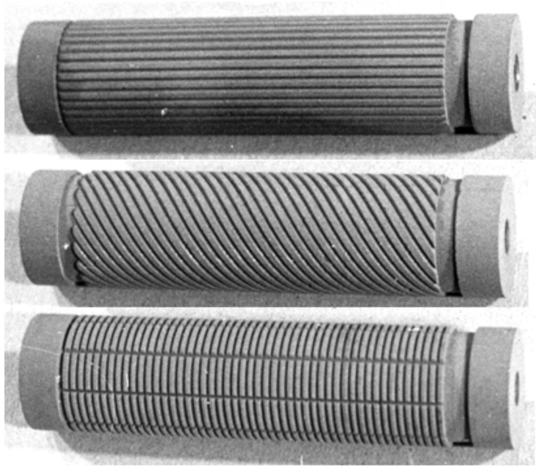


Fig. 36: Schematics of CPL (left) and LHP (right)

LHP, CPL: High-Performance Sintered Metal Wicks





Specifications

Material	Nickel	Titanium	Copper
Eff. pore radius, µm	0.5 - 2	3 - 10	2 - 20
Porosity, %	65 - 75	55 - 70	50 - 75
Permeability, m^2	~ 10^{-14}	~ 10^{-13}	~ 10^{-13}

Fig. 37: High performance wicks for LHP

Separate Type Heat Pipe (Loop Thermosyphon)

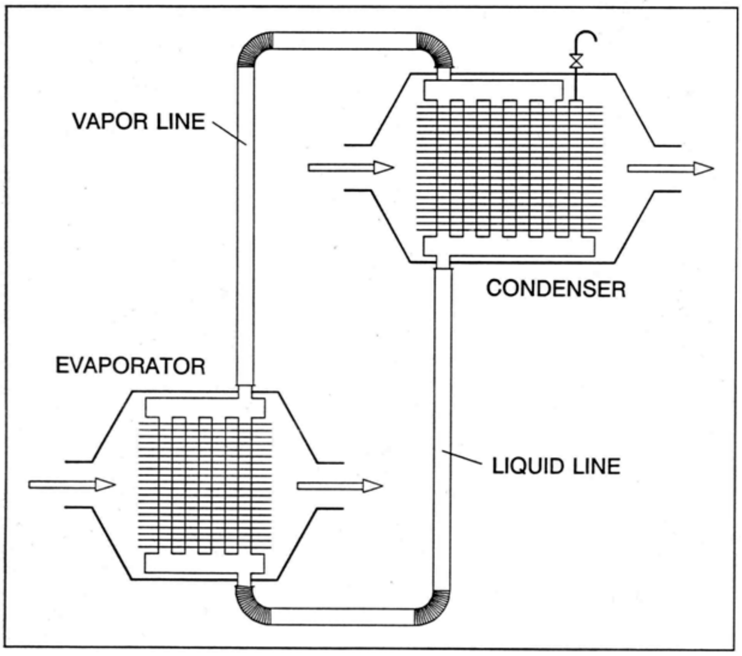


Fig. 38: Schematic of separate type heat pipe (loop TS)

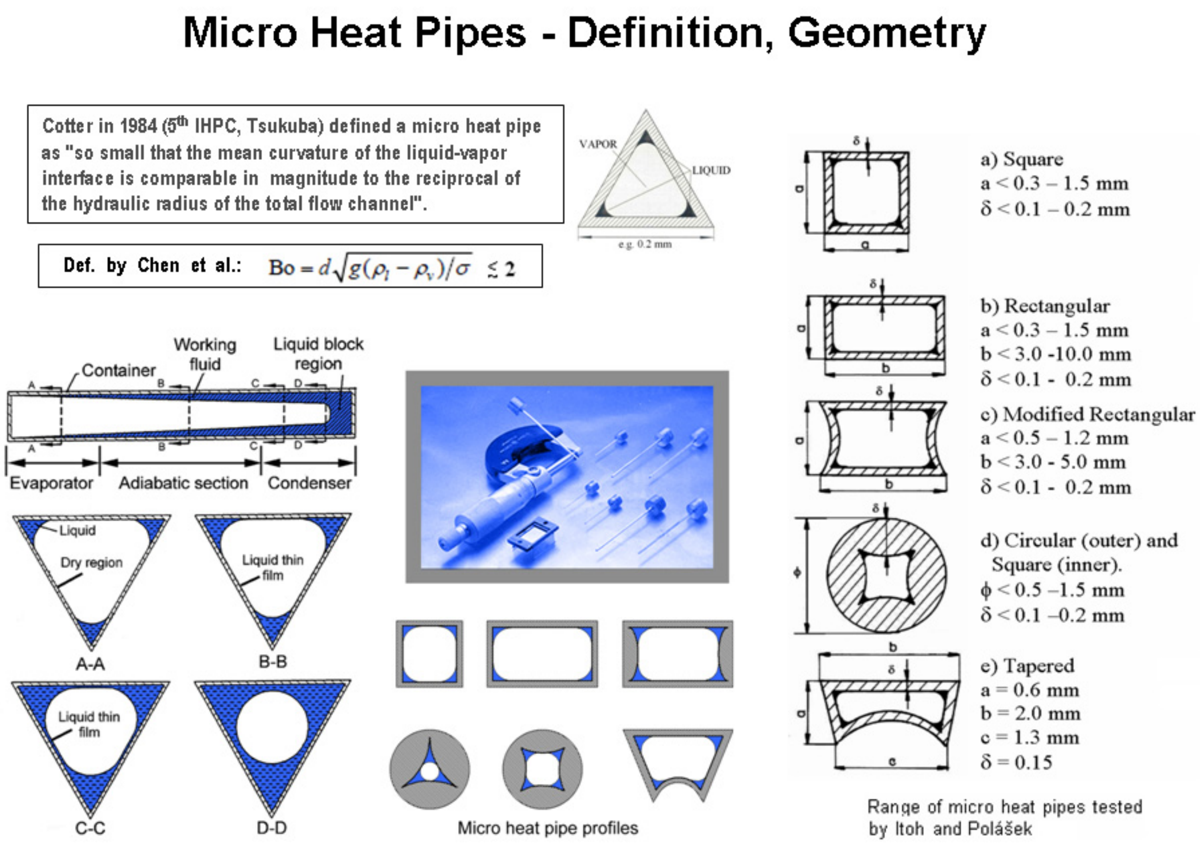


Fig. 39: Micro heat pipe: definition, geometries

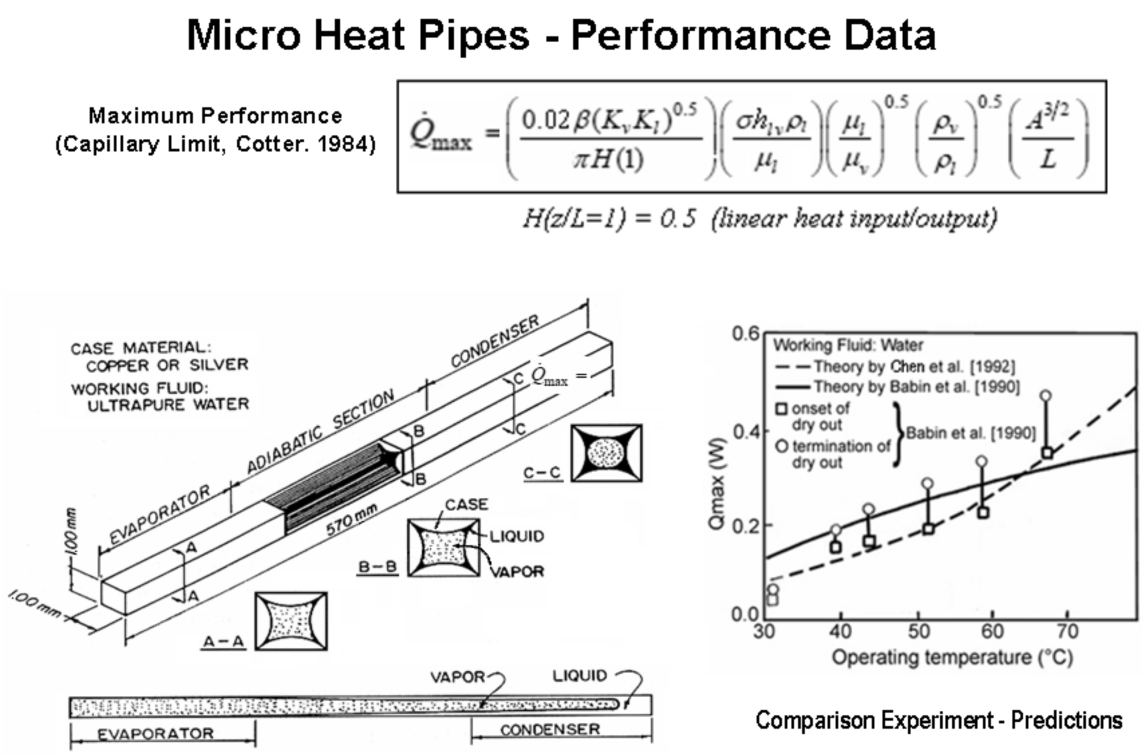


Fig. 40: Micro heat pipe: performance data

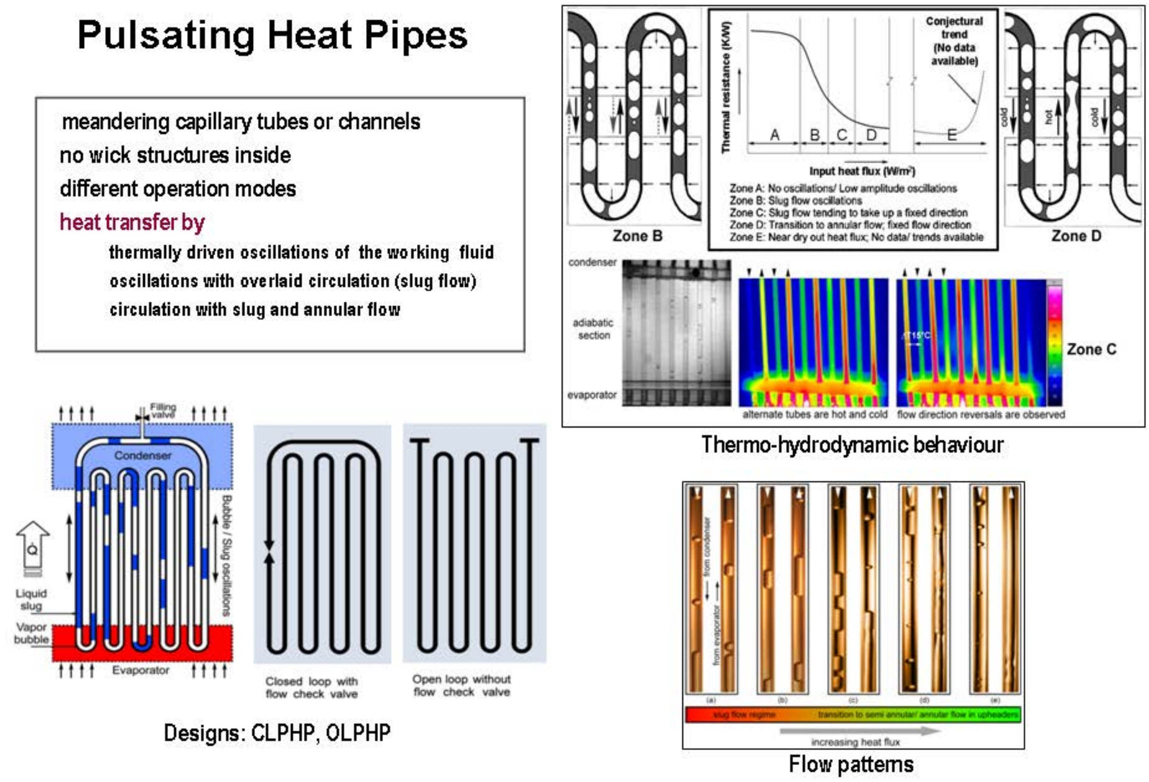


Fig. 41: Pulsating heat pipes: operation principle, designs, thermo-hydraulic behaviour of CLPHP

Passive Liquid-Vapour Phase Change Heat Transfer Devices

External Force Field Driven Devices

- **Gravity driven:**
  - **Two-phase Thermosyphons (TS)**
  - ❑ Open two-phase TS
  - ❑ Closed two-phase TS
    - Without flow separator
    - With flow separator
      - – Single tube TS
      - – Loop TS (Separate type TS)
  - ❑ Supercritical HP (single phase)
  - ❑ Anti gravity TS with bubble pump (vapour lift pump), auxiliary power
    - Single stage
    - Multi stage
- **Centrifugal field driven**
  - ❑ Rotating heat pipe
- **Electric Field driven**
  - ❑ Electro-hydrodynamic HP
  - ❑ Electro-osmotic HP
- **Magnetic Field driven**
  - ❑ Magnetic fluid HP

Internal Force Field Driven Devices (1)

- **Surface Tension driven:**
  - **Heat Pipes (HP)**
  - ❑ Constant conductance HP (CCHP)
  - ❑ Variable conductance HP (VCHP)
    - Non feed back controlled (T_V = const.)
      - – Gas control
        - + Cold wicked reservoir
        - + Hot non-wicked reservoir
        - + Gas absorption reservoir
      - – Liquid control (T_C = const.)
        - + Constant liquid reservoir volume
        - + Variable liquid reservoir volume
    - Feedback controlled (T_{HS} = const.)
      - – Gas control
        - + Passive feedback control
        - + Active feedback control (auxiliary power)
      - – Vapour flow control (active)
      - – Liquid control (active and passive)
  - ❑ HP diode
    - Liquid control
      - – Liquid trap
      - – Liquid blockage

Internal Force Field Driven Devices (2)

- Gas control
- Capillary pressure control
- Gravity control (TS)
- ❑ HP thermal switch
- ❑ HP triode
- ❑ Micro HP
- ❑ Capillary pumped loop
- ❑ Loop heat pipe (LHP)
- **Pressure Gradient driven:**
  - ❑ Pulsating / oscillating HP (PHP / OHP)
    - Open loop
    - Closed loop
      - – Without check valve
      - – With check valve
  - ❑ Periodic / intermittent operating HP (anti-gravity TS with "passive pumping module"(PPM))
- **Concentration Gradient driven:**
  - ❑ Osmotic HP

Fig. 42: Classification of passive liquid-vapour phase-change heat transfer devices

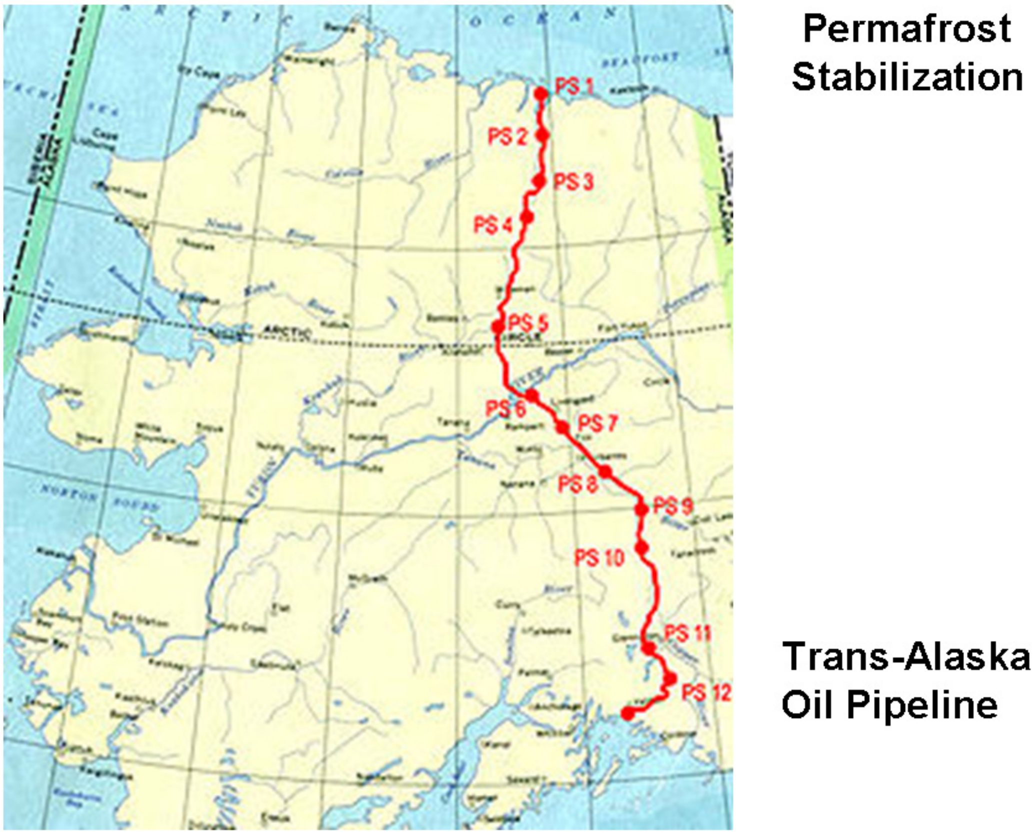


Fig. 43: Permafrost stabilization: Trans-Alaska oil pipeline

Fig. 44: Permafrost stabilization: details of Trans-Alaska oil pipeline

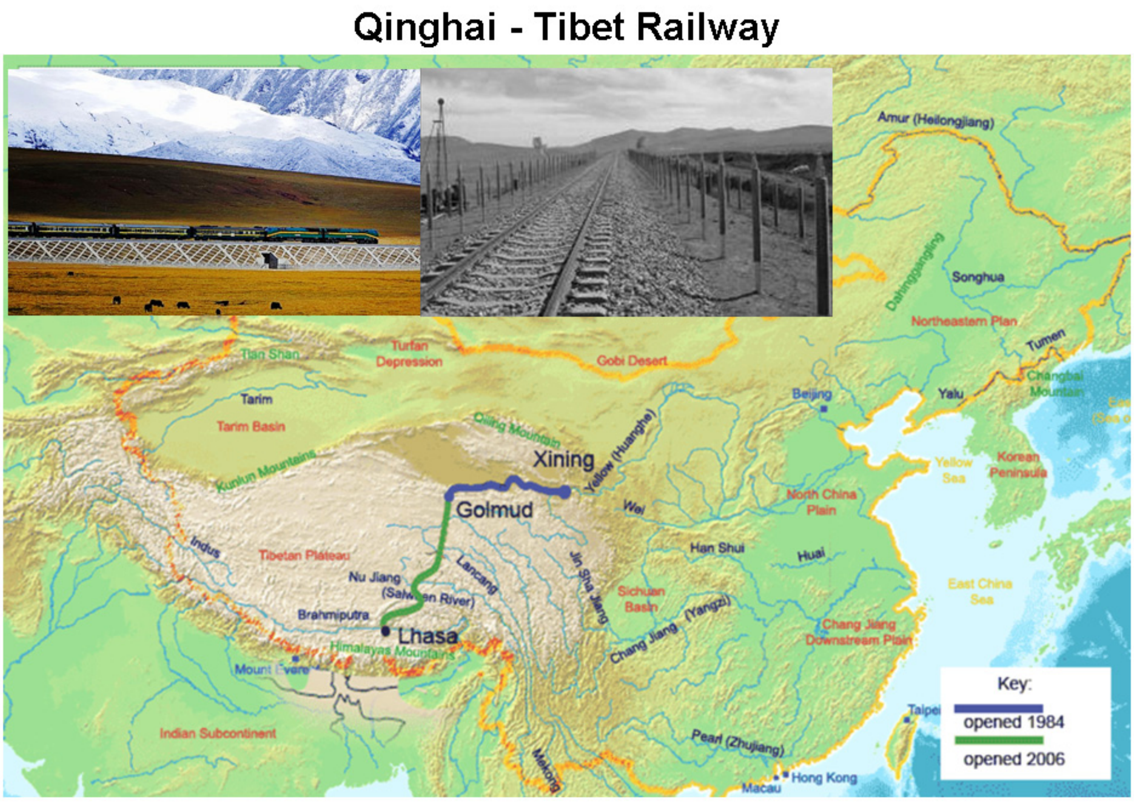


Fig. 45: Permafrost stabilization: Qinghai-Tibet railway

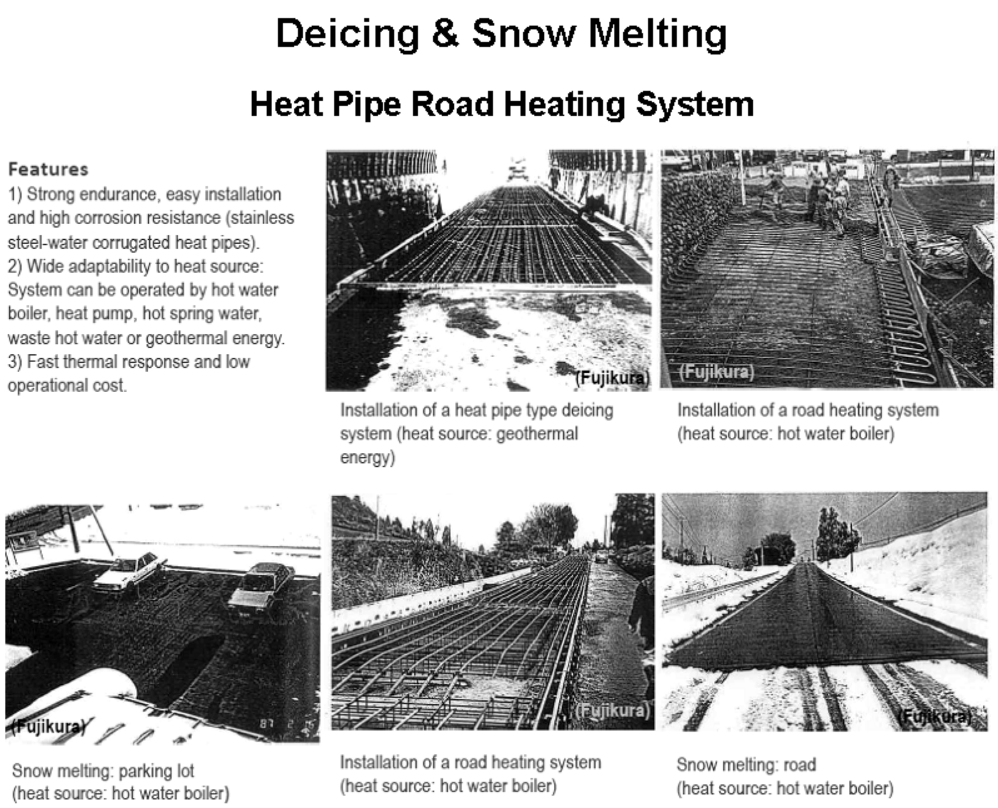
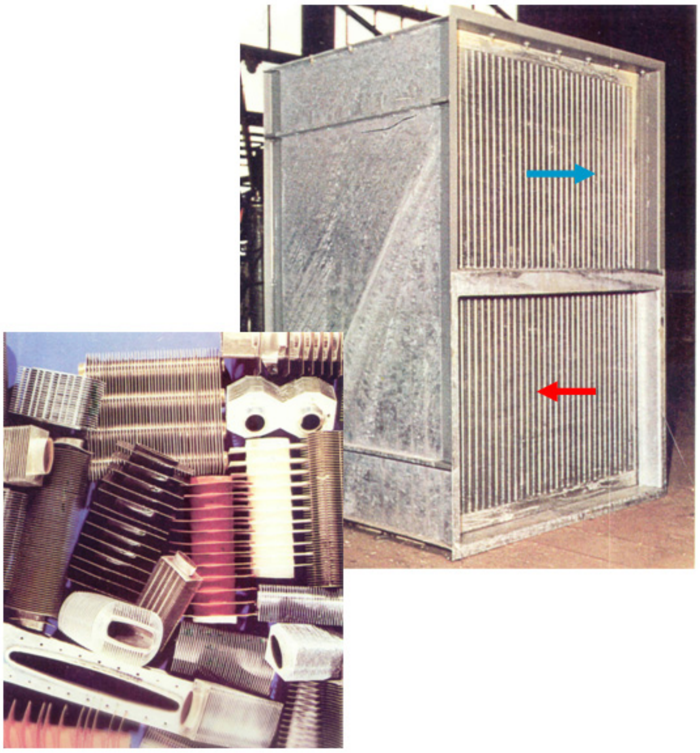


Fig. 46: Deicing of roads in Japan

Heat Recovery / Heat Exchangers

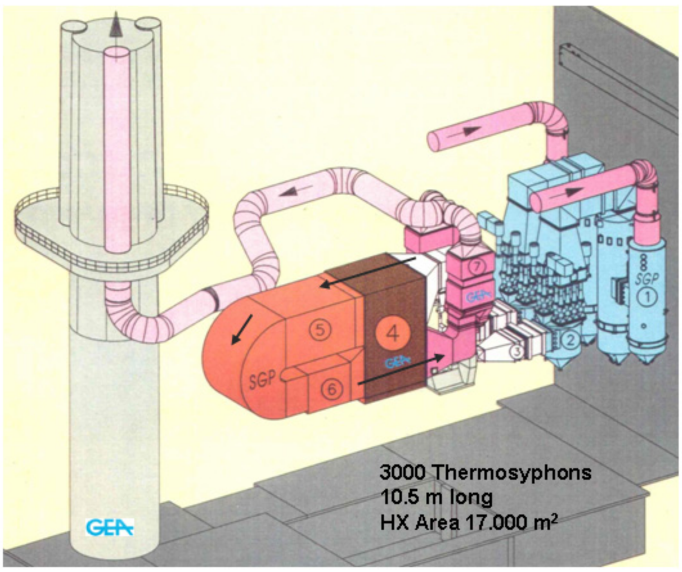


Advantages of Heat Pipe Heat Exchangers

- **compact design**
  (high heat fluxes possible, dense finning on both gas sides possible)
- **very high redundancy**
- **isothermal elements**
  (no cold areas, temperature nowhere below acid dew point; no local corrosion)
- **gas - tight partition wall**
  (no cross - mixing / contamination of gas flows)
- **passive system, no moving part**
  (no auxiliary energy consumption)
- **little maintenance**
- **high flexibility with respect to design and implementation**
  (separate type heat pipe systems)
- **cost-effective series production of the heat pipes and easy assembly of the heat exchanger**

Fig. 47: “Heat pipe” heat exchanger and samples of finned HPHX tubes; advantages of HPHX

HP-HX System in Incineration Plant



① Flue gas desulphurization
② Droplet eliminator
③ Gas drier
④ Heat pipe heat exchanger

⑤ Afterburner
⑥ Catalyst
⑦ Heat recovery for district heating system)

Individual Heat Pipe Modules

GEA ECOSTAT heat-pipe system
ECOSTAT-modules carbon-steel type

Fig. 48: HPHX system in incineration plant: schematic; heat pipe modules

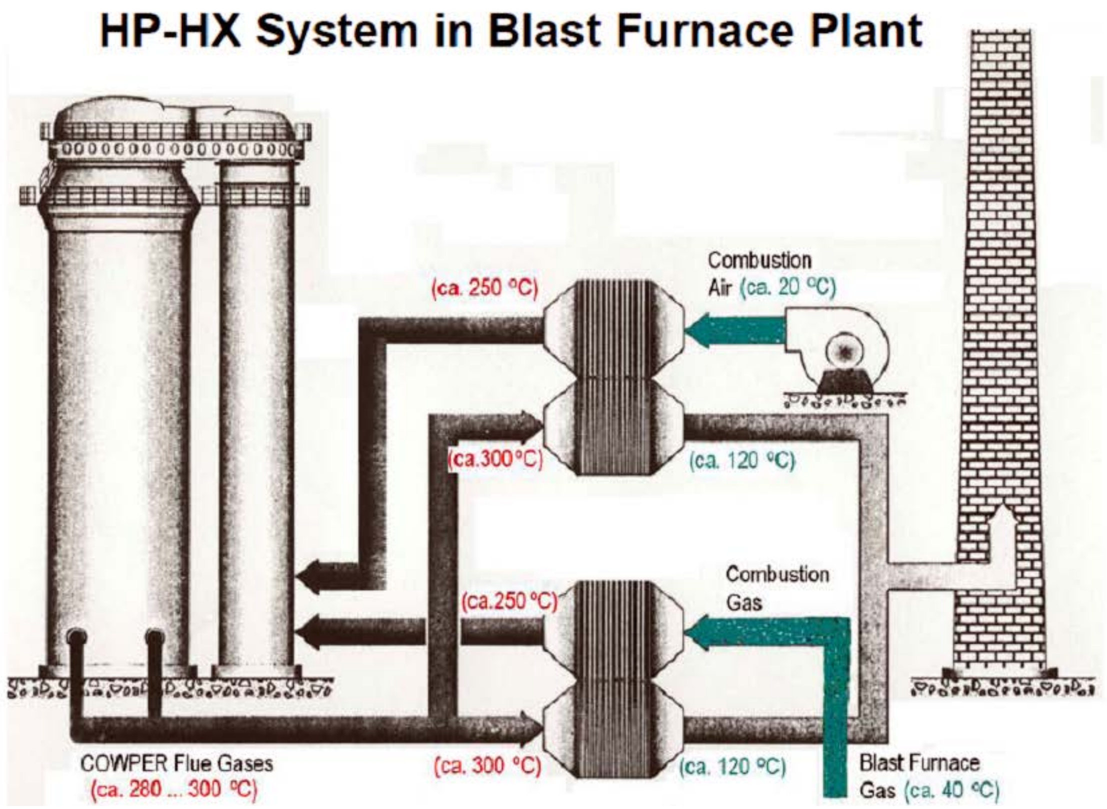


Fig. 49: HPHX system in blast furnace plant. Schematic

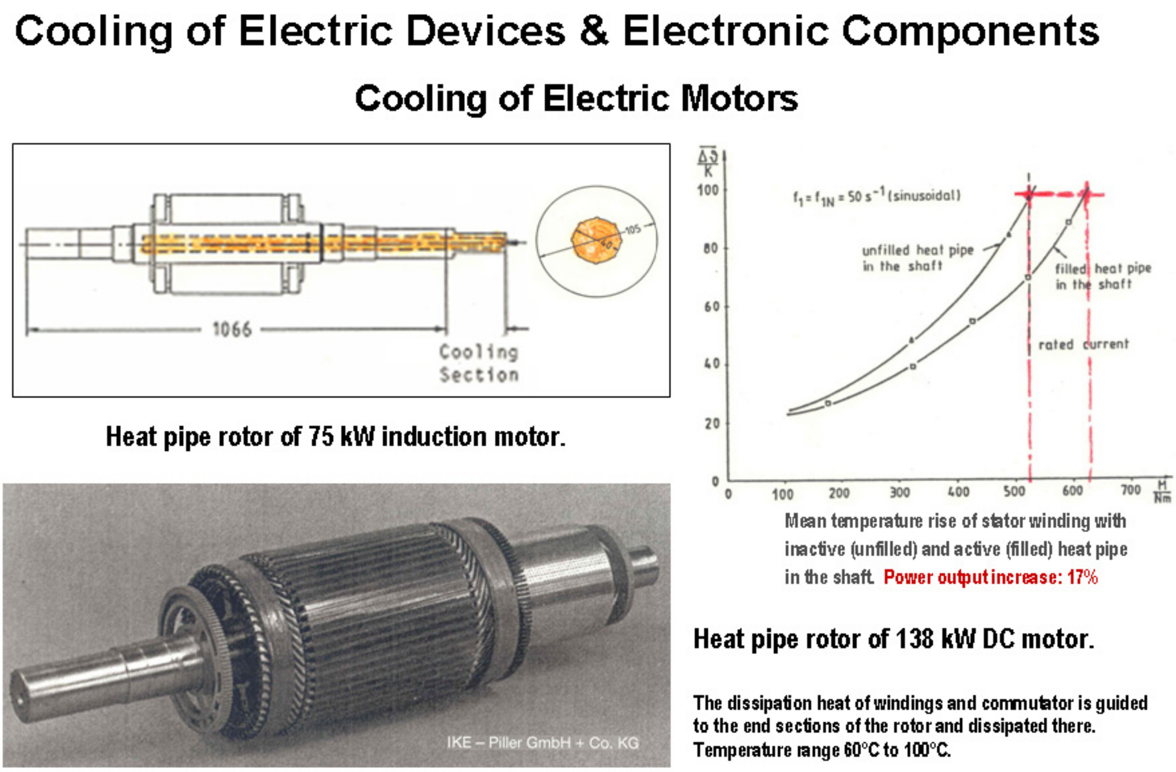


Fig. 50: Heat pipe cooled electric motor: view, schematic, performance increase

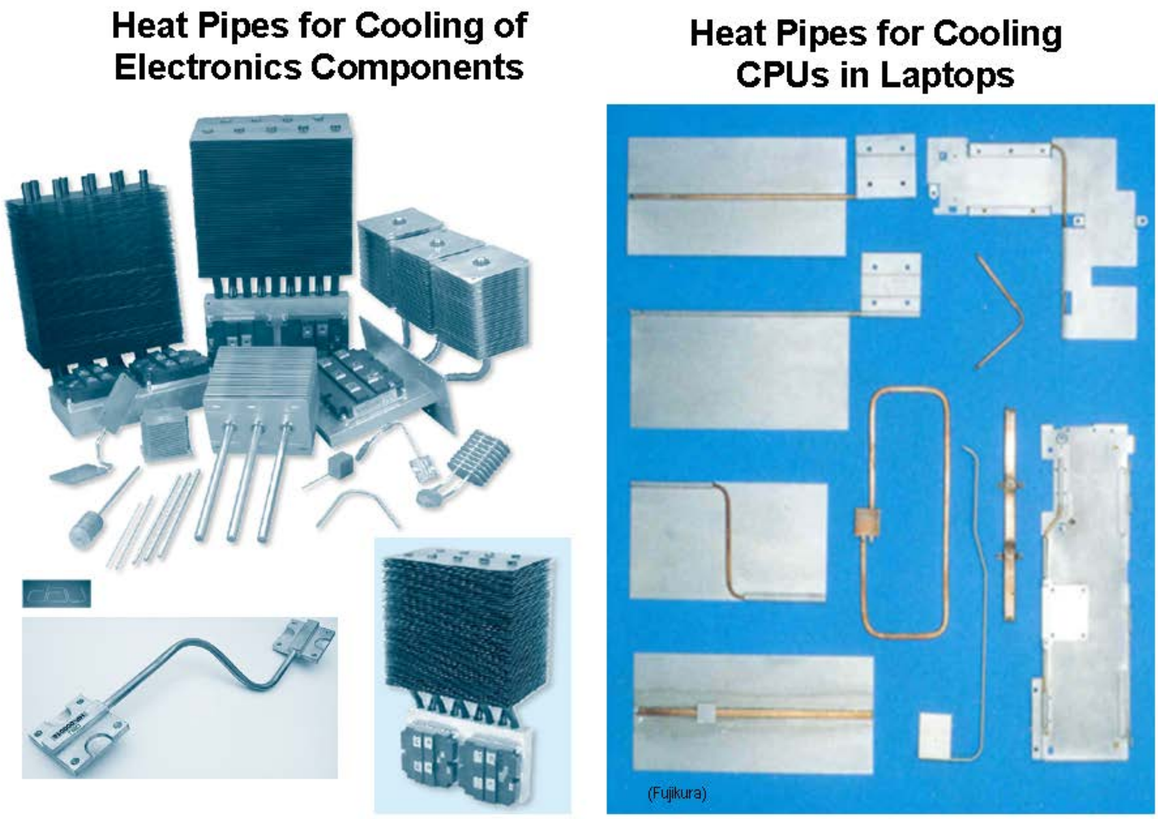


Fig. 51: Heat pipes for cooling of electronic components

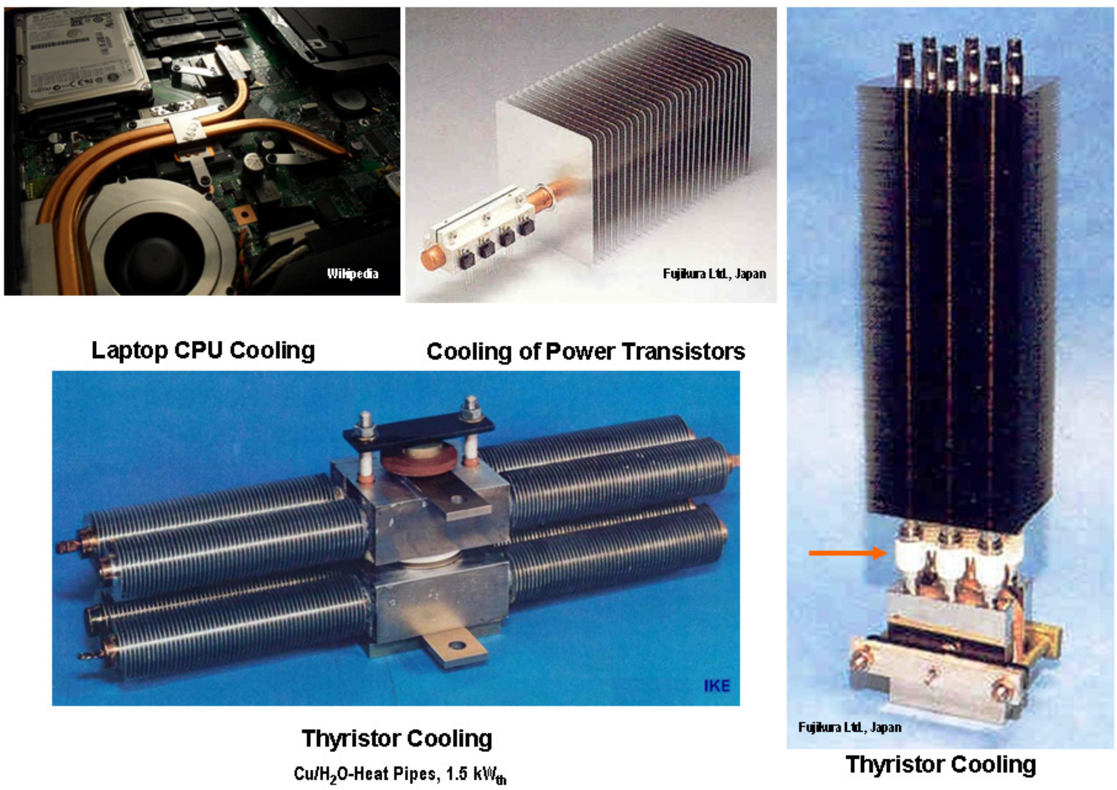


Fig. 52: Heat pipes for cooling of CPU, transistors, thyristors

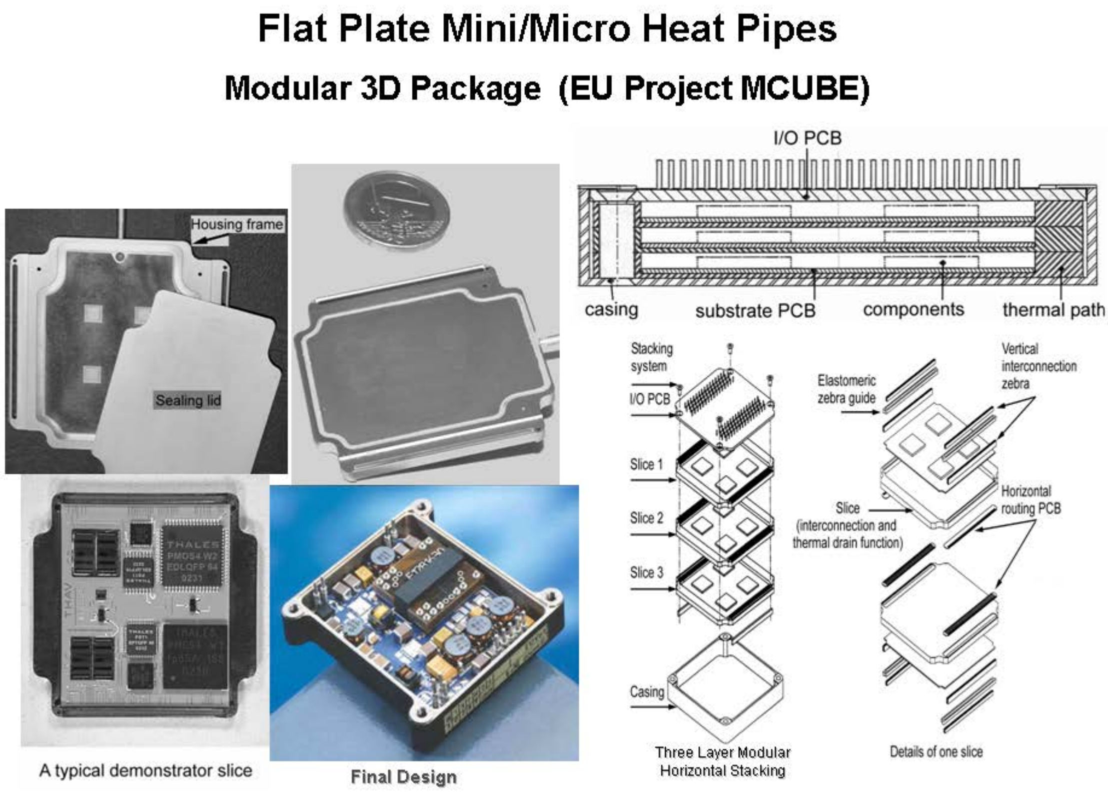


Fig. 53: Heat pipe cooled modular 3D package (EU Project MCUBE)

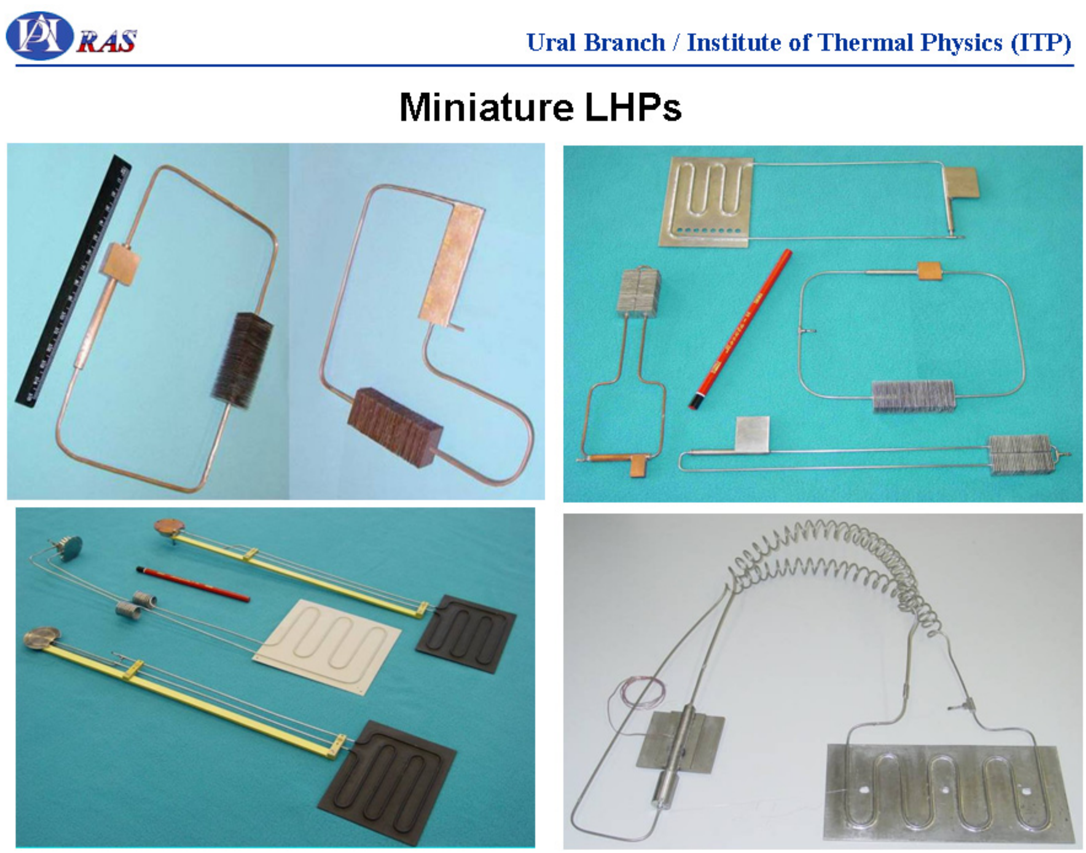


Fig. 54: Miniature LHP

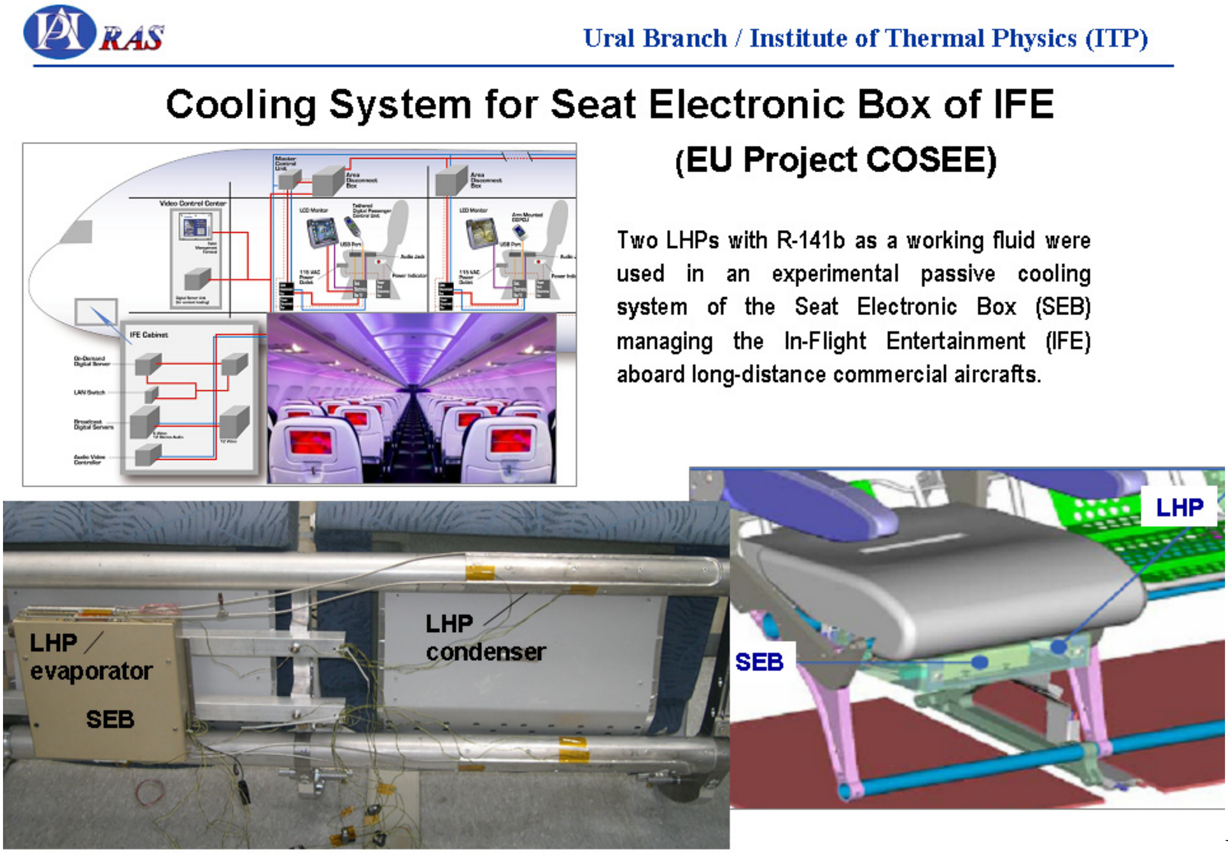


Fig. 55: LHP for cooling of seat electronic box in commercial aircraft (EU Project COSEE)

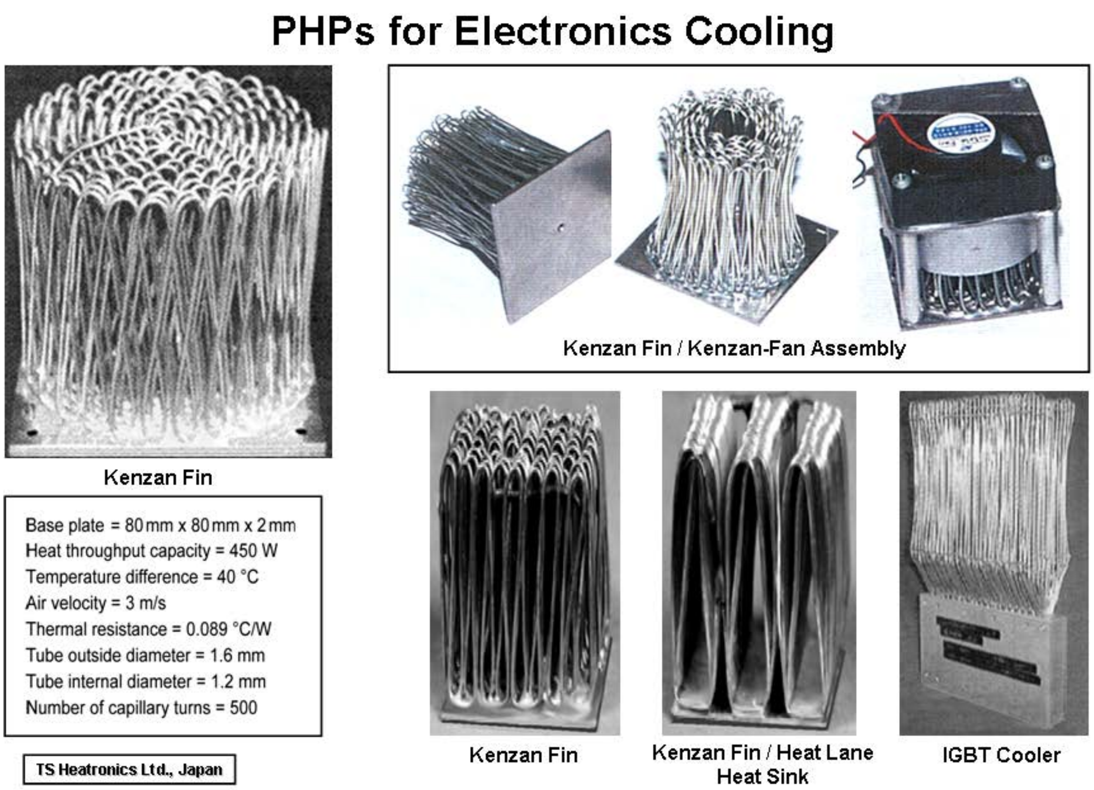


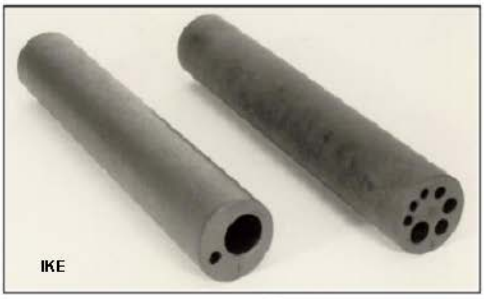
Fig. 56: PHP for electronics cooling

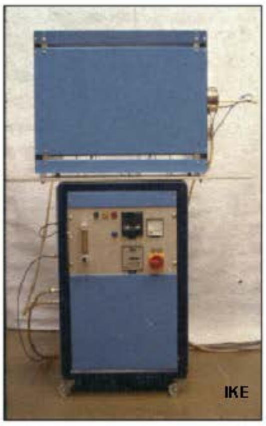
Liquid Metal Heat Pipes for Calibration Purposes

Heat Pipe Black Bodies

cavity diameters up to 50 mm
emissivity > 0.999

$-50°C \leq T \leq 50°C$:	NH_4
$50°C \leq T \leq 250°C$:	H_2O
$300°C \leq T \leq 660°C$:	Cs
$550°C \leq T \leq 1100°C$:	Na
$1100°C \leq T \leq 1600°C$:	Li



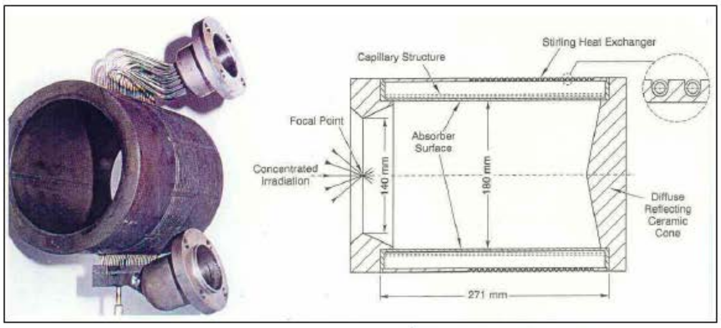


Au - Fixpoint Blackbody

temperature stability	$\pm$ 0.1 K
isothermality	$\pm$ 0.05 K
cavity emissivity	0.999 +

Fig. 57: Sodium heat pipe black bodies for calibration purposes

Liquid Metal Heat Pipes for Solar Receivers



2nd Generation Na-Heat Pipe Receiver (IKE/DLR)

Solar Dish Power Station with Heat Pipe Receiver / Stirling Engine

Fig. 58: Sodium heat pipe receiver for solar dish power station

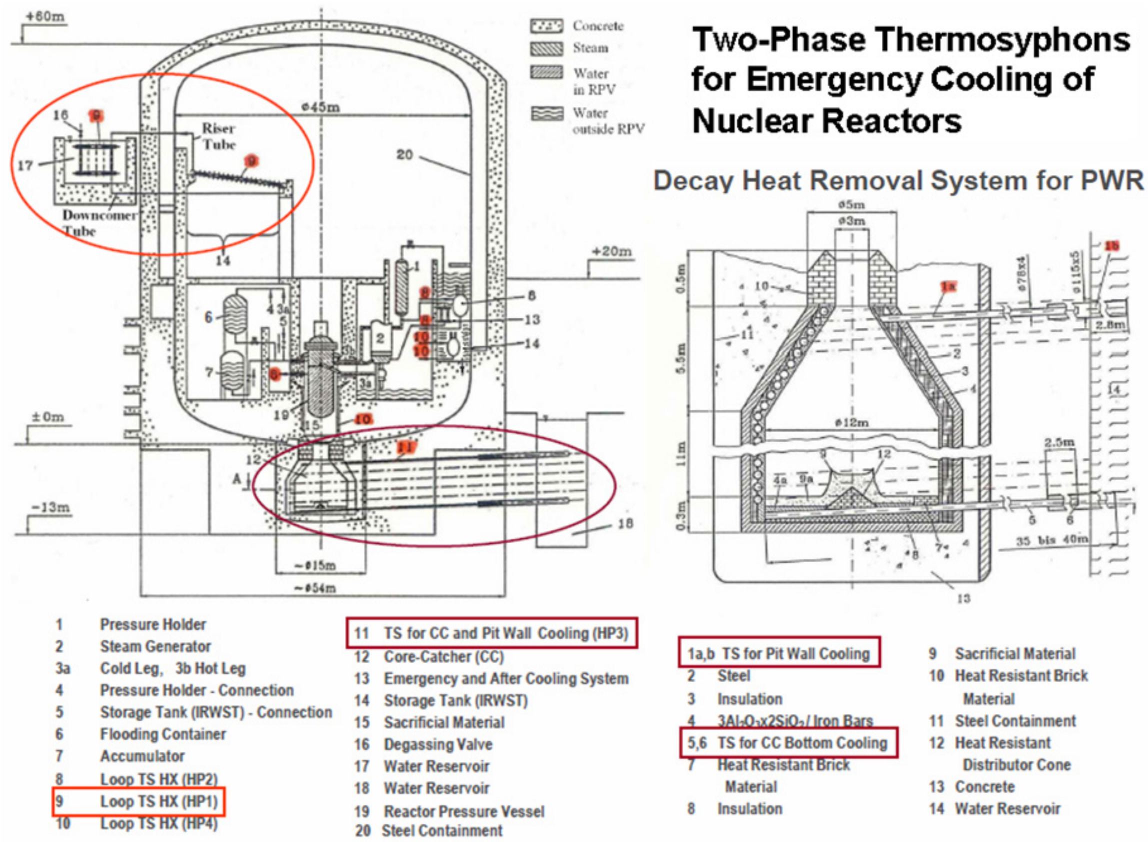


Fig. 59: Various heat pipe/thermosyphon systems for decay heat removal from a pressurized water reactor

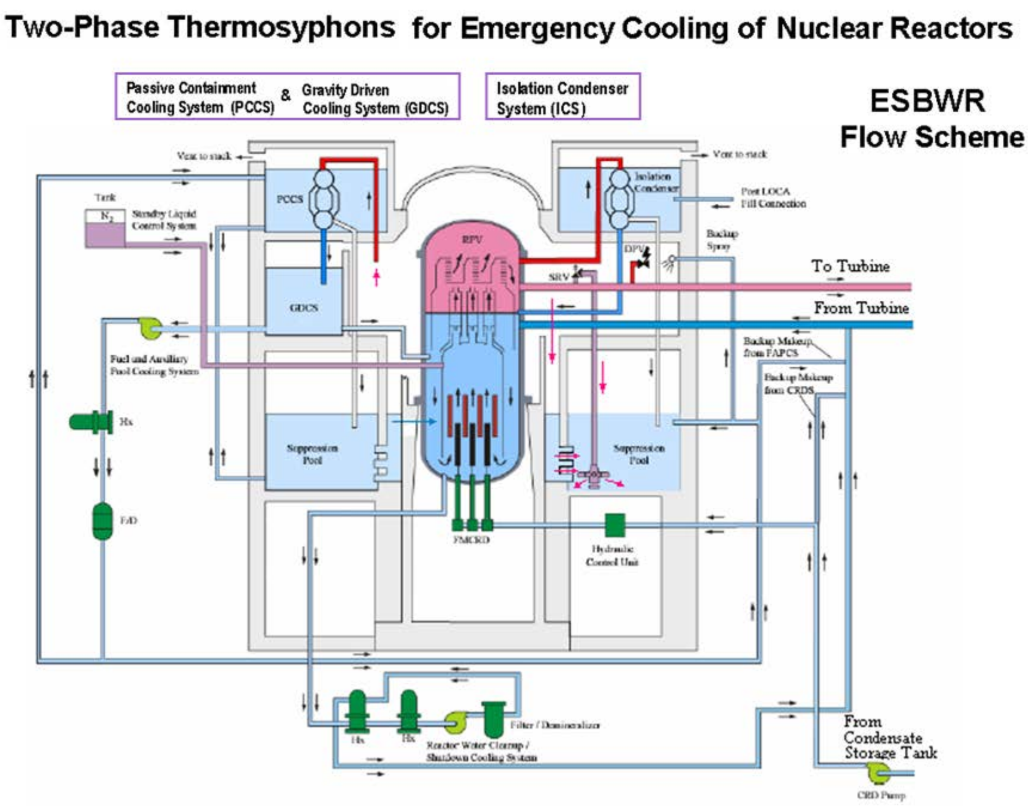


Fig. 60: Emergency cooling systems of ESBWR: PCCS/GDCS and ICS

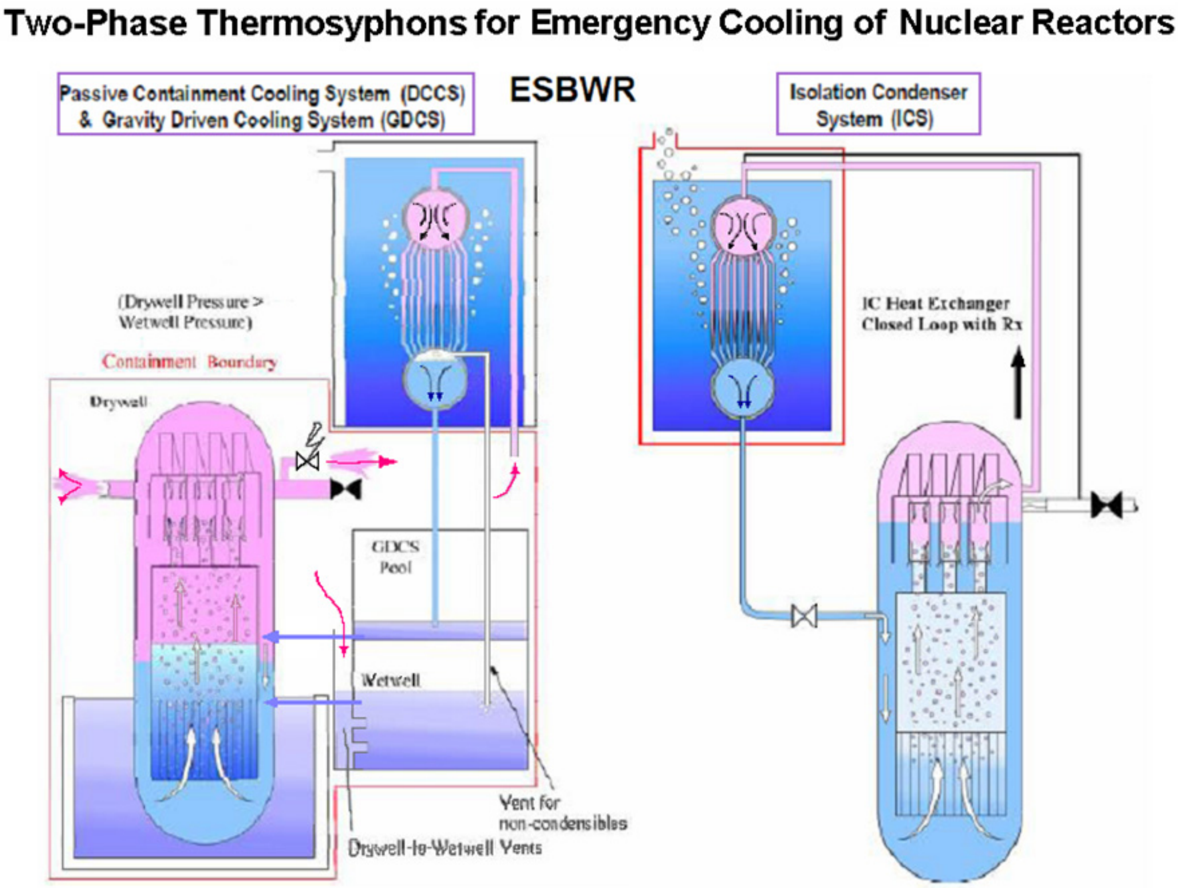


Fig. 61: Emergency cooling systems of ESBWR: PCCS/GDCS and ICS