14 Heat Pipes

R. C. Prager,* M. Nikitkin,† and B. Cullimore‡

Overview

Heat pipes use a closed two-phase liquid-flow cycle with an evaporator and a condenser to transport relatively large quantities of heat from one location to another without electrical power. A heat pipe can create isothermal surfaces; as a thermal "transformer," it can change the flux density of a heat flow; and it can function in various ways as a thermal-control device. One-way (diode) heat pipes have been tested and flown, as have variable-conductance heat pipes (VCHPs), which maintain a constant-temperature evaporator surface under varying load conditions. Because the driving mechanism in a heat pipe is capillary pumping, a relatively weak force that is provided by a wick, the pipe may be susceptible to severe performance degradation when operating in a gravitational field. Planning is therefore needed to facilitate the ground testing of systems that include heat pipes.

How a Heat Pipe Works

Consider a simple horizontal heat pipe in equilibrium with an isothermal environment. The liquid in the wick and the vapor in the vapor space are at saturation. If heat is applied to the evaporator, raising its temperature, liquid in the wick evaporates (removing some of the added heat), which depresses the meniscus in the evaporator because less liquid remains there. This process also raises the local vapor pressure, because that pressure must be in saturation with the heated liquid in the wick.

The difference between the increased curvature of the meniscus in the evaporator wick and the unchanged meniscus in the condenser wick causes a difference in capillary pressure sufficient to pull liquid from the condenser wick toward the evaporator wick. This action replenishes the liquid in the evaporator wick. At the same time, heated vapor flows from the evaporator to the condenser, which is at a lower pressure. When this vapor comes in contact with the cooler surfaces of the condenser, it condenses. This cycle of evaporation and condensation is shown schematically in Fig. 14.1.

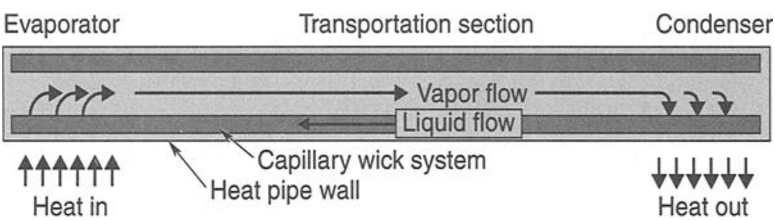

Fig. 14.1. Heat-pipe schematic.

*The Aerospace Corporation, El Segundo, California.
†Swales Aerospace, Beltsville, Maryland.
‡C&R Technologies, Littleton, Colorado.

Because the latent heat of vaporization of most heat-pipe working fluids is high, only small amounts of fluid need to flow to transport significant quantities of heat. The driving mechanism, the temperature difference between the evaporator wall and the condenser wall, is also small.

Types of Heat Pipe

Constant-Conductance Heat Pipe

This most basic heat pipe consists of a working fluid, a wick structure, and an envelope. This pipe is used to move heat from one location to another (possibly changing the flux density in the process) or to isothermalize a surface. It need not be shaped like a conventional cylindrical pipe—flat plates several feet across have been built and tested as heat pipes for special applications. Constant-conductance heat pipes are often categorized according to the type of wick structure they use.

Groove Wicks

The simplest heat-pipe wick design consists of axial grooves in the wall of extruded aluminum tubing. Grooves can be formed in tubes of other materials, such as copper (by swaging) or even refractory metals (by deposition), but they are formed most often in tubes of aluminum. This class of wick is very susceptible to gravitational effects during ground testing, but it is relatively inexpensive to produce and it performs very consistently. Its moderate heat-transfer capability is sufficient for many applications. Most grooves are rectangular or trapezoidal, but some have more complex shapes, such as the “teardrop” or “keyhole,” which can be extruded with difficulty (Fig. 14.2).

“Monogroove” Design

The monogroove design, a high-capacity design consisting typically of a wick in one large, teardrop-shaped groove connected to a vapor space (Fig. 14.3), can be considered an extension of the basic groove concept. Unlike a heat pipe with many

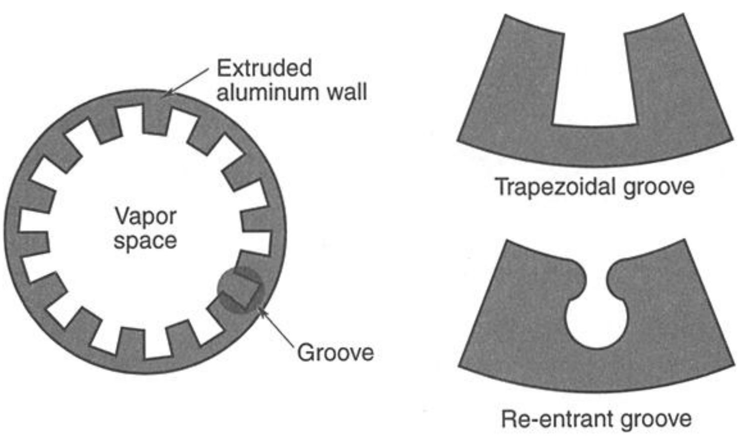

Fig. 14.2. Grooved heat pipe.

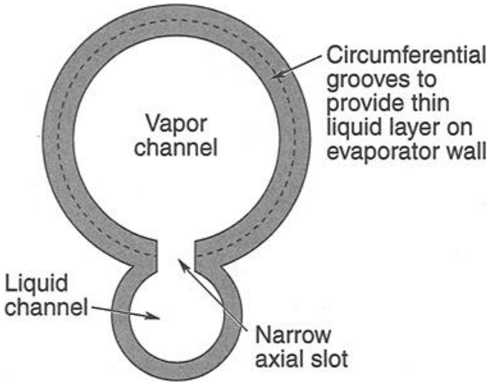

Fig. 14.3. Monogroove heat pipe.

smaller grooves of the same total area, the monogroove heat pipe has a large single groove that provides relatively unrestricted longitudinal flow. Liquid is distributed on the evaporator wall by means of a secondary wick consisting of small circumferential grooves or screen. This design has shown very high capacity during ground testing, but it encountered difficulties during early shuttle testing. Later experiments were more successful. As of this writing, no monogroove heat pipe has been used on a production spacecraft.

Composite Wicks

Among composite wicks, the simplest (and the oldest heat-pipe wick) consists of several layers of screen fastened to the inside wall of a heat pipe. More capacity can be obtained by using more layers of screen, to increase the wick flow area—at the cost of increasing the heat-pipe temperature difference resulting from the temperature drop needed to conduct heat through the thick saturated wick. To overcome this penalty, some heat-pipe manufacturers separate the wick into two parts, the portion that spreads the fluid circumferentially about the wall of the evaporator, and the portion that carries the fluid down the length of the heat pipe. The former, kept as thin as possible, can consist of circumferential grooves cut in the wall of the heat pipe or of a single layer of screen or metal mesh bonded to the wall. The latter is held off the wall by means of legs or straps, or makes contact with the wall in only a few places. This type of wick has capacities similar to the axially grooved heat pipe, but has much more capability when tilted. Because the wick must be assembled of relatively fragile materials, care is required in building such a pipe, and no two supposedly identical pipes will perform in exactly the same manner. Sample wick designs of this type are shown in Fig. 14.4.

Artery and Tunnel Wicks

This class of heat pipe is based on the composite wick, but provides one or more relatively unrestricted liquid-flow paths in parallel with the longitudinal wick. These paths will fill with fluid in space, because of minimum surface-energy considerations, and will greatly reduce the viscous pressure drop in the heat pipe, thereby increasing capacity. When properly designed, these arteries will fill as the

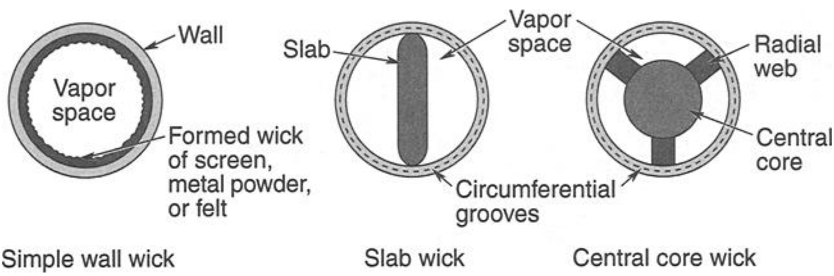

Fig. 14.4. Composite wicks.

heat pipes operate in a gravitational field. Wicks in this class can be blocked by bubbles of noncondensable gas in the arteries (see Abhat *et al.*[14.1] and Saaski[14.2]), but they are attractive because of their large heat-transfer capability in a small envelope. If the liquid in the artery remains subcooled when it reaches the evaporator, bubble formation can be avoided. A number of mechanical schemes have been proposed and tested to prevent bubbles from blocking the arteries of VCHPs (see Eninger[14.3]). These pipes are particularly prone to bubble formation because the liquid in the artery contains dissolved control gas, which tends to come out of solution as the liquid warms during its transit of the pipe from condenser to evaporator. Cross sections of some of these wick structures are shown in Fig. 14.5.

Diode Heat Pipes

A constant-conductance heat pipe can be modified so that operation occurs normally in one direction but ceases when an attempt is made to transfer heat in the other, "wrong" direction, resulting in a diode action. Even when blocked, however, the pipe transfers some heat, if only by conduction down the pipe wick and

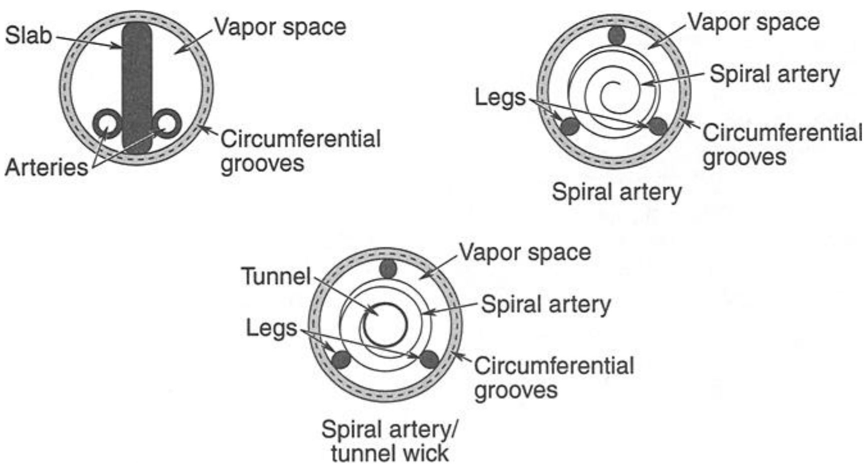

Fig. 14.5. Artery and tunnel wicks.

wall. This type of heat leak is particularly significant in cryogenic systems. Common diode heat pipes are the liquid-trap, liquid-blockage, and gas-blockage diodes.

Liquid-Trap Diode

The most common type of heat-pipe diode, the liquid-trap diode has a wicked reservoir at the evaporator end designed so that it is heated by the same environment that heats the evaporator. Although the envelopes are connected, the reservoir wick is not connected to the rest of the heat pipe. When, during normal operation, heat is applied to the evaporator and reservoir, heat is transferred from the evaporator to the condenser as in the constant-conductance heat pipe, and any fluid in the reservoir wick evaporates and joins the vapor flow to the condenser. (The reservoir wick should be dry during normal operation.) When ends of this pipe are reversed, and the evaporator and reservoir become cooler than the condenser, some of the hot vapor coming from the condenser condenses in the reservoir and is lost to the rest of the heat pipe. Sufficient liquid is tied up in the reservoir to cause the pipe to dry out. "Shutoff" is neither instantaneous nor complete. A schematic of the operation of this type of diode is shown in Fig. 14.6.

Liquid-Blockage Diode

At its condenser end, the liquid-blockage diode (Fig. 14.7) has a wicked reservoir cooled by the same environment that cools the condenser. The reservoir's wick is not in contact with that of the remainder of the heat pipe, and it is normally full of

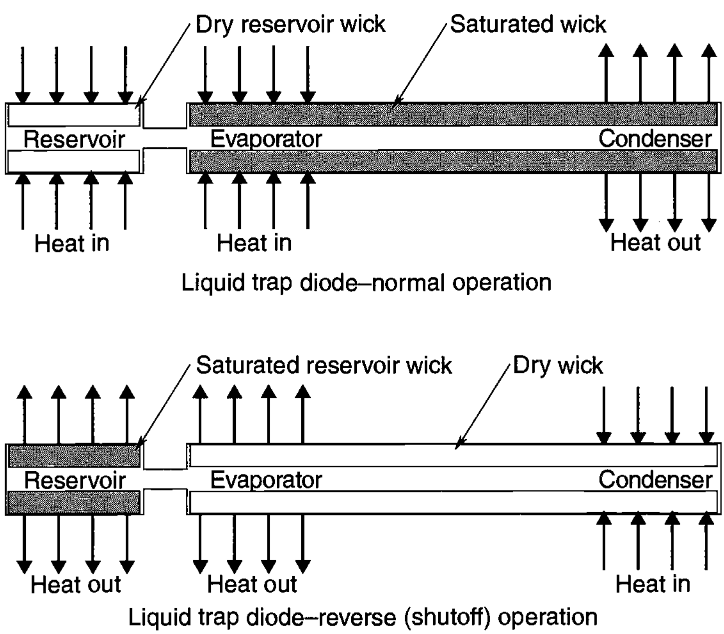

Fig. 14.6. Liquid-trap diode.

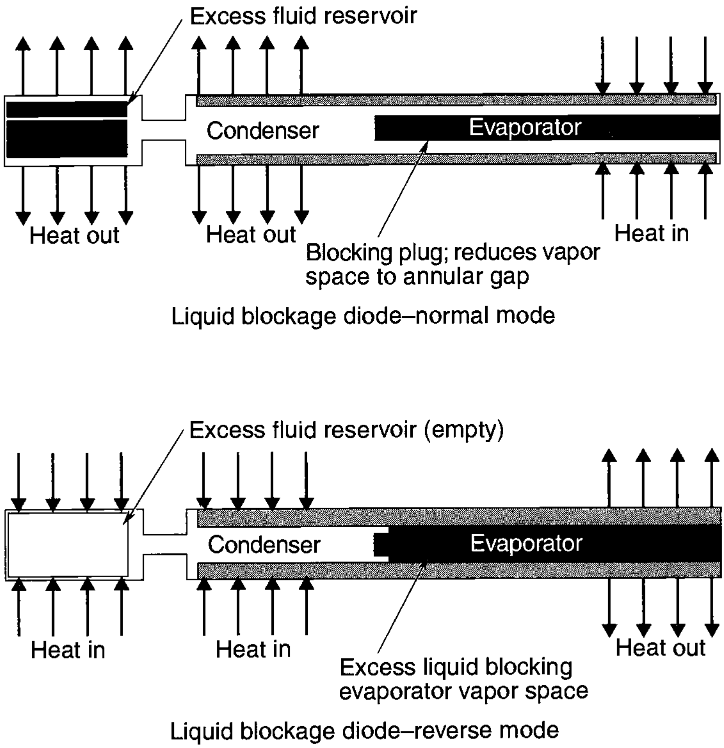

Fig. 14.7. Liquid-blockage diode.

working fluid—in effect, it traps a large fluid slug. When the ends of the pipe are reversed, the fluid slug travels to the normal evaporator end, where it completely fills the evaporator vapor space (and that of a large portion of the transport section), preventing condensation. Optimum design of the wick structure and vapor space must be compromised to control the liquid slug during shutoff, and such control requires maintaining close tolerances during the manufacturing process. Proper control of the fluid (and therefore operation of the diode) in a gravitational field requires maintaining the gap between the evaporator wall and the blocking plug at a size that enables the gap to fill with liquid if it is available.

Gas-Blockage Diode

The gas-blockage diode is similar in design to the liquid-blockage diode, except the reservoir, which can be unwicked, contains a noncondensable gas. When the ends of the pipe are reversed, the gas flows to the evaporator and, as above, completely fills the vapor space, preventing condensation. However, as the temperature rises, the gas slug can be compressed to the point where the heat pipe will start working again. Furthermore, convection within the gas slug may be a significant heat-leak component. A schematic of the operation of this type of diode is shown in Fig. 14.8.

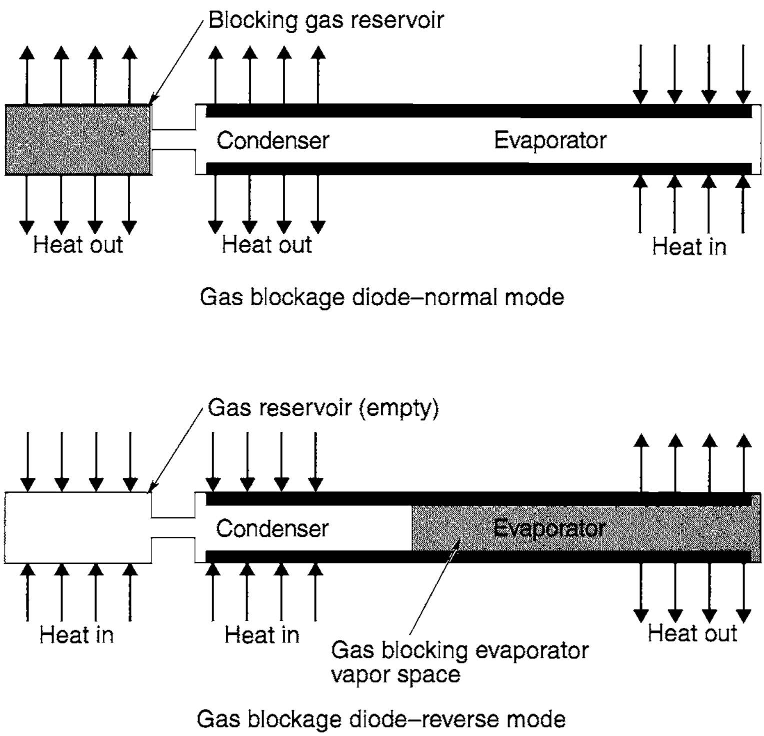

Fig. 14.8. Gas-blockage diode.

Other Diodes

Any heat pipe that has a wick with a finer pore size in the evaporator than in the condenser or the adiabatic section will show some signs of diode operation, and its capacity will differ depending upon the direction in which it is trying to move heat. The most extreme case is that of a heat pipe with no wick in the condenser (see the capillary pumped loop [CPL], below), as the pipe will dry out quickly and shut off if heat is applied there.

VCHPs

VCHPs use a gas reservoir connected to the end of the condenser. The reservoir is filled with a noncondensable gas to control the operating area of the condenser based on the evaporator temperature. (In effect, in a typical spacecraft application, the active radiator area becomes a function of the electronics-box cold-plate temperature, with increasing box temperatures leading to increased radiator areas.) Although complicated models of the gas front exist, the gas front may be considered an impermeable floating piston. If the temperature at the cold plate rises, the vapor in the evaporator (at the saturation pressure of the liquid in the evaporator) rises rapidly. The pressure of the mixture of control gas and vapor in the reservoir must rise to compensate, so the "gas-front-as-piston" will move further into the condenser, decreasing the volume of control gas. This process, shown schematically in Fig. 14.9, opens up more of the condenser area to heat-pipe operation. A number of VCHP schemes have been flown; they have differed mainly in how they treat the reservoir. Some have wicks, some are kept hot or cold by exposure to different environments, and some become elements of what is arguably an active thermal-control system by means of heaters connected via feedback control to sensors at the evaporator or payload. Sufficient control gas is usually present in the reservoir to enable these pipes to function as gas diodes if the ends of the heat pipe are reversed. The VCHP operation temperature profile in Fig. 14.10 shows temperature as a function of position along the pipe.

Hybrid (Mechanically Assisted) Systems

Hybrid systems are essentially extensions of the CPL. They cannot be considered passive thermal control systems, because of the addition of small pumps to force liquid flow. Because they are two-phase systems, only small quantities of the working fluid need to be carried to the evaporation site in the liquid phase to transport large amounts of heat energy. Several such systems had been proposed for use on the Space Station, and a number of prototypes have been built and tested.

Analysis

Heat-Pipe Capacity (Capillary Pumping Limit)

Return flow of liquid from the condenser to the evaporator is caused by differences in the capillary pressure between the evaporator and condenser. The capillary

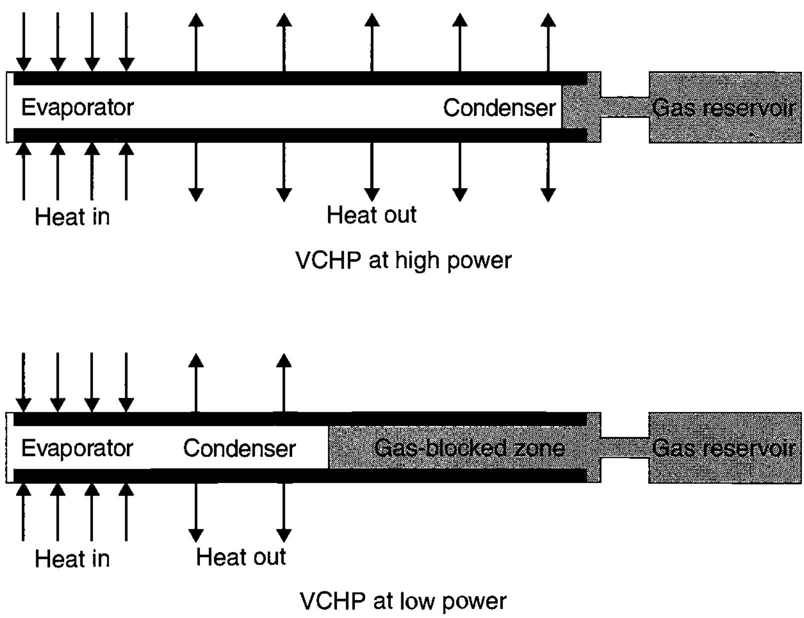

Fig. 14.9. VCHP operation schematic.

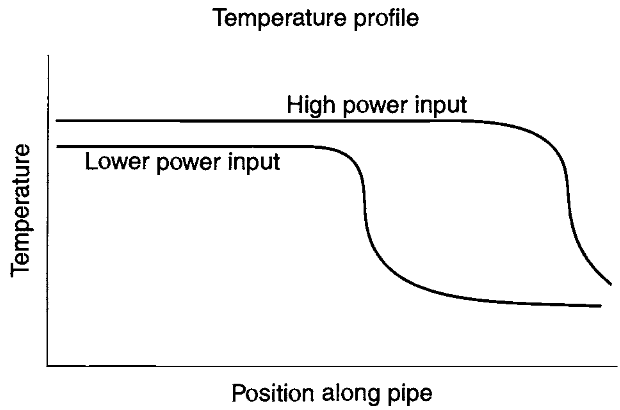

Fig. 14.10. VCHP operation temperature profile

pressure acting on the liquid surface is inversely proportional to the radius of curvature of the fluid surface at the liquid/vapor interface in the wick. For purposes of the analysis, the liquid surface in the condenser is usually assumed to be flat, so that the radius of curvature (and hence the capillary force) is zero. As liquid evaporates, the meniscus in the evaporator depresses, causing a difference in capillary pressure between the evaporator and condenser surfaces (Fig. 14.11). This difference in pressure pulls liquid through the wick from the condenser to the evaporator in an attempt to restore equilibrium.

A heat pipe "dries out" when the flow of working fluid through the wick caused by this pressure difference is insufficient to supply liquid at the same rate at which working fluid is being vaporized in the evaporator. This point is illustrated in the Eq. (14.1), which balances the pressure drops in the system:

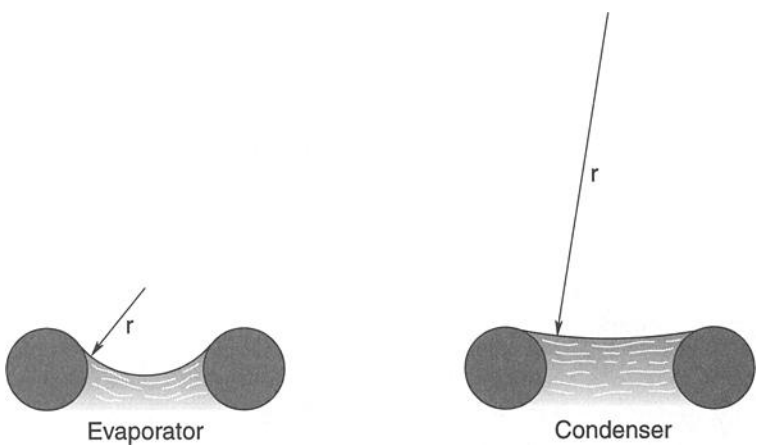

Fig. 14.11. Depression of the meniscus.

$$\Delta P_{CAPILLARY} - \Delta P_{GRAVITY} = \Delta P_{LIQUID} + \Delta P_{VAPOR}. \tag{14.1}$$

In this equation, $\Delta P_{CAPILLARY}$ (capillary pressure rise) is the maximum possible difference in capillary pressure between the evaporator and the condenser. This term is a function of the surface tension (which depends on the choice of working fluid and the temperature) and the wick pore size (which depends upon the wick material and type of wick).

$\Delta P_{GRAVITY}$ (gravity head loss) is the "head loss" that must be overcome by capillary pressure to sustain fluid in the evaporator. In addition to gravity, other accelerations, such as those on a spinning spacecraft, affect the value of this term.

ΔP_{LIQUID} (liquid pressure drop) is the pressure loss resulting from viscous flow through the wick. This term is simple for an axial-groove wick, but it can become extremely complicated for a composite-artery wick, where viscous pressure losses in liquid flowing through complicated structures of layered screens, metal felt, or sintered powder must be modeled. Expressions for these losses usually contain empirical constants, which is one of the reasons why performance testing of each pipe is usually necessary.

ΔP_{VAPOR} (vapor pressure drop) is the pressure loss resulting from vapor flow from the evaporator to the condenser. This term is usually small unless the vapor density is very low or the vapor velocity is high because of constricted vapor space.

The exact equation will depend upon the wick design used. Many formulations are given in the references.

Thermodynamic Considerations

If operation near the freezing point is needed—as would be the case for water at typical room temperatures, for almost any cryogenic liquid, or for liquid metals at start-up—high vapor velocities and large vapor-pressure drops will be encountered, because in these situations the vapor density and pressure are very low. These large pressure drops cause their own temperature drops in the pipe (because saturation temperature is a function of pressure). In some cases, the pressure drop in the vapor required to support the calculated heat-pipe capacity would result in a negative vapor pressure in the condenser, an obvious impossibility. Under similar low-density conditions, choked flow (the "sonic limit") has been observed in liquid-metal heat pipes. Although not a true limit, the operating temperature of the heat pipe rises so thermal equilibrium can be established, which may cause the temperature to rise beyond the desired range. In short, do not design a heat pipe that must run in a temperature regime where its working fluid has a very low vapor pressure.

If the relative velocity of liquid and vapor is high enough (as measured by the Weber number), liquid can be pulled out of the wick and returned to the condenser in the form of droplets entrained in the vapor. This phenomenon (the "entrainment limit") was first observed in liquid-metal heat pipes where the droplets could be heard to "ping" against the end cap. It is an operating limit in that, to support a given rate of heat transfer from the evaporator, an excess of liquid must be pulled through the wick, because not all of the liquid will reach the evaporator.

The "boiling limit" or "heat-flux limit" is concerned with the flux density of the thermal load on the evaporator. Even if the heat-pipe wick could theoretically return the liquid from the condenser required by the heat load, if the load is concentrated in too small an area, nucleate boiling can occur in the evaporator wick. The creation of bubbles in an otherwise filled wick reduces the area of the wick available for fluid flow, and hence reduces the capacity of the wick.

Working Fluids

The choice of working fluid is usually governed by the temperatures of the desired operating range. A heat-pipe working fluid can be used effectively between a temperature somewhat above its triple point and another that is below its critical temperature. If the triple point is approached too closely, temperature drops in the vapor flow increase (see the discussion above, "Thermodynamic Considerations"). As the critical point is approached, the distinction between liquid and vapor blurs, and the surface tension drops to zero. (The pressure that must be contained by the envelope also increases significantly.) The triple points and critical temperatures of several heat-pipe working fluids are given in Table 14.1.

Table 14.1. Heat-Pipe Working Fluids[a]

	Melting Point		Boiling Point		Critical Temp.	
Fluid	(K)	(°F)	(K)	(°F)	(K)	(°F)
Hydrogen	14.0	–434.4	20.4	–423.0	33.0	–400.3
Neon	24.5	–415.6	27.1	–410.9	44.4	–379.8
Oxygen	54.3	–361.8	90.2	–297.3	154.8	–181.1
Nitrogen	63.1	–346.0	77.3	–320.4	126.2	–232.4
Ethane	89.9	–297.8	184.5	–127.6	305.5	90.2
Methane	90.7	–296.4	111.4	–259.2	190.5	–116.8
Methanol	175.2	–144.3	337.9	148.5	513.2	464.1
Acetone	180.0	–135.7	329.4	133.2	508.2	455.1
Ammonia	195.5	–107.8	239.8	–28.0	405.6	270.4
Water	273.2	32.0	373.2	212.0	647.3	705.4
Potassium	336.4	145.8	1032.2	1398.3	2250.0	3590.0
Sodium	371.0	208.1	1152.2	1614.3	2500.0	4040.0
Lithium	453.7	357.0	1615.0	2447.0	3800.0	6380.0

[a]Data from Brennan and Kroliczek, *Heat Pipe Design Handbook*.[14.4]

Two parameters have been developed to aid in comparing the relative performance of heat-pipe working fluids. The first, the "zero-g figure of merit," is given by $\frac{\sigma\rho\lambda}{\mu}$, where σ is the surface tension, ρ is the liquid density, λ is the latent heat of vaporization, and μ is the dynamic viscosity. This parameter neglects vapor flow entirely, but for most applications, vapor flow is not the limiting factor. The group of fluid properties included in the parameter definition appears in the heat-pipe capacity equation. A second parameter, the "one-g figure of merit" or "wicking height factor," compares the relative sensitivity to gravity effects of working fluids: $\frac{\sigma}{\rho}$, where the properties are as defined above. It is a relative measure of how high a given wick structure will be able to pump a working fluid in a gravitational field (or as a result of inertia effects, as in a spinning spacecraft).

Material Compatibility

Because a heat pipe is a completely sealed container, any chemical reactions between the working fluid and the wall or wick material can be disastrous. None of the reaction products can escape, and any material that is consumed cannot be replaced. Certain combinations of materials, such as ammonia and copper, are known to react quickly with one another, and hence are not likely to be chosen, even by a novice.

However, combinations of materials that are traditional and acceptable in the chemical-process industry (such as water and stainless steel, or water and nickel) have been demonstrated to react with one another, generating noncondensable gas. In general, the cryogenic working fluids up through ammonia can be used with either stainless steel or aluminum (although some evidence indicates that ammonia reacts slowly with aluminum, and the combination of ammonia, aluminum [as is found in a wall material], and stainless steel [such as would be found in a typical wick material] can react more quickly with one another).

Methanol works well with stainless steel but reacts with aluminum. Water seems to work well with copper, and possibly monel, but not with 304 or 316 stainless steel or nickel. Some short-term success has been achieved with carbon steel, but pipes using it appear to be generating hydrogen gas, which diffuses through the pipe wall; this observation indicates an internal reaction is taking place.

Materials available for higher-temperature (liquid-metal) heat pipes must hold together at those higher temperatures and be inert to some very corrosive working fluids. This area is still under investigation.

Testing

During Fabrication

The heat-pipe envelope will be checked for leaks during the fabrication process, usually with a helium mass-spectrometer leak detector. However, once the pipe is sealed at the fill tube, the integrity of this seal is open to question. Although some chemical tests have been used (see Edelstein[14.5]), the most thorough seems to be checking for the presence of working fluid outside the heat pipe when it is placed in an evacuated chamber.

Performance of each heat pipe as a function of tilt should be measured at some typical operating temperature(s) to determine whether the wick functions properly. Testing at a low temperature will show whether noncondensable gas is present. (At high temperatures, the noncondensable gas can be compressed into a thin plug so that it isn't detectable using thermocouples mounted on the heat pipe.)

If the heat pipe is to be installed in a spacecraft in a position where it will be tested vertically (with gravity assist) during system-level testing, such as a thermal-vacuum or thermal-balance test, then it must be tested in the same orientation with a similar heat load before installation. In this way, the performance of the heat pipe that will be seen in the vacuum chamber will be known before the test is performed. This data will help to avoid unpleasant surprises and scrambling for logical explanations at a time when the heat pipe can't be reached without breaking vacuum and tearing open the spacecraft.

In the case of a heat pipe that is to be curved in three dimensions and can't be tested in a single plane, some manufacturers build a test pipe with the same number of curves in the wick, but with all of the curves in a single plane. In this way, the wick performance to be expected in space can be characterized.

After Integration into the System

After integration of a heat pipe into a system, the heat pipe should be verified to determine whether any deterioration took place during the integration procedure, and also to verify the performance of the integrated thermal control system.

Heat-Pipe Applications and Performance

The most obvious application of a heat pipe is one requiring physical separation of the heat source and sink. If a heat pipe is used, all hardware to be cooled need not be mounted directly on radiator panels, and relatively inefficient conductive couplings need not be used. (Requirements for this type of coupling are usually found in cases where boxes must be cooled and kept close to each other for more efficient electrical or microwave design.) By the same token, heaters need not be mounted directly on hardware to be heated if a heat pipe is employed.

A closely related class of applications is that of the thermal transformer. In this scenario, a small high-powered box is mounted on one side of a radiator with integral heat pipes; the heat generated is spread and dissipated at a much lower flux density over the entire surface of the radiator. This approach also permits more efficient use of available "real estate"—the area available for a radiator is seldom centered symmetrically about the heat source, facing the optimal direction.

Heat pipes have been used to reduce temperature gradients in structures to minimize thermal distortion. The telescope tube of the NASA Orbiting Astronomical Observatory (OAO) had three ring-shaped heat pipes to minimize circumferential temperature gradients. The ammonia heat pipes worked throughout the eight years of mission life.

The diode heat pipe was first proposed as a means of connecting a device to two radiator panels on opposite sides of a spacecraft, with the understanding that at least one of the radiators would be free of any direct solar load at all times during the orbit. The diodes would couple the device to the cold radiator, while preventing heat from leaking back into the system from the radiator in the sun. This type of thermal-design problem, in which heat from a temporarily warm radiator or from a failed refrigerator must be kept from leaking back into the system, is an obvious application for a diode heat pipe.

The VCHP can control the amount of active radiator area, providing reasonably good temperature control without the use of heaters. This capability is particularly attractive if electrical power is limited, and this type of design has been flown on a number of satellite experiments. However, if the application requires maintaining a box or baseplate at a virtually constant temperature, feedback control (at the expense of some heater power) may be employed. A sensor on the baseplate of the device to be controlled can be routed to an onboard computer, and whenever the temperature drops below the desirable range, heaters on the VCHP reservoirs are activated, causing the control gas to expand and block off more of the radiator area. If the temperature rises above the range desired, power to the reservoir heaters is reduced, increasing the active radiator area. This concept usually requires less power than the direct use of heaters on the box or system to be controlled.

The use of flexible heat pipes or rotatable joints in heat pipes to cool devices on rotating or gimbaled platforms has been proposed, but flexible heat pipes tend to have too much resistance to motion, and rotating joints in heat-pipe walls leak under extreme conditions. These areas are still under active investigation.

Heat-Pipe References

More detailed discussions of a broad range of topics concerning heat-pipe design and applications can be found in Refs. 14.1 through 14.17. In addition, papers concerning new developments in heat-pipe design and analysis and discussing new applications, or the results of tests or experiments, are usually presented at the AIAA Thermophysics Conference. Volumes of proceedings from the International Heat Pipe Conference, which is held every four years, can be found in technical libraries.

LHPs and CPLs

Because of performance advantages, unique operational features, and recent successful flight experiments, the Western-heritage capillary pumped loops (CPLs) and the Russian-heritage loop heat pipes (LHPs) are rapidly gaining acceptance in the aerospace community. They are used as baseline thermal-control technology for a number of missions, including NASA's EOS-AM, GLAS, SWIFT, and GOES; ESA's ATLID; CNES's STENTOR; a retrofit mission for the Hubble Space Telescope (HST); and various commercial geosynchronous communication satellites.

Despite wide use of the emerging CPL and LHP technologies, fundamental confusion persists about their operation, limitations, and even their similarities and differences. This discussion, by engineers who have participated in CPL and LHP development on both sides of the Atlantic, explains the concepts behind them for potential users.

Initially many perceived CPLs and LHPs as alternatives to conventional heat pipes at high transport powers (> 500 W, with up to 24 kW demonstrated), but in recent years the intrinsic advantages of a small-diameter piping system without distributed wick structures have been exploited at low powers (20 to 100 W).

Many advantages of CPLs and LHPs are only truly exploited when these devices are considered early in the design phase, rather than treated as replacements for existing heat-pipe–based designs. Their advantages include:

- tolerance of large adverse tilts (a heat source up to 5 m above a heat sink, facilitating ground testing and even enabling many terrestrial applications)
- tolerance of complicated layouts and tortuous transport paths
- easy accommodation of flexible sections, make/break joints, and vibration isolation
- fast and strong diode action
- straightforward application in either fixed-conductance or variable-conductance (active-temperature-control) mode
- separation of heat-acquisition and -rejection components for independent optimization of heat transfer footprints and even integral independent bonding of those components into larger structures
- accommodation of mechanical pumps
- apparent tolerance of large amounts of noncondensable gases, which means an extended lifetime
- no vapor/liquid entrainment concerns or boiling limits

For this discussion, only single-evaporator systems such as those currently being baselined will be described. Multiple-evaporator as well as multicondenser systems are under active development by many parties, and they already have flight heritage in a flight experiment. However, to introduce the plethora of design options possible in those systems would cause unnecessary confusion in this basic discussion of LHPs, CPLs, and their advantages and disadvantages. Likewise, complications that can arise when multiple LHPs are networked together will not be addressed here.

CPL Overview

CPLs were invented in the United States in the 1960s, but active development on them did not begin until around 1980. Through most of the 1980s, NASA Goddard Space Flight Center (GSFC) sponsored the majority of CPL development with OAO Thermal Systems and Dynatherm (both now part of Swales Thermal Systems) performing a large part of the development and test effort. A typical single-evaporator CPL system, such as that used for several instruments on the EOS-AM spacecraft, is depicted in Fig. 14.12.

In this system, vapor generated by the evaporator flows to the condenser, where not only is it condensed, but also a certain amount of subcooling (5 to 10°C minimum) is generated. The liquid flows back toward the evaporator, whereupon it enters the core (the inner diameter) via an optional bayonet. One purpose of the bayonet is to position any gas or vapor voids nearest the coldest (incoming) liquid such that they are minimized. Often secondary wicks or arteries are positioned within the liquid core in an attempt to prevent any bubbles from axially blocking off portions of the wick, especially in microgravity environments. The liquid is pulled radially through the primary wick, and it is vaporized on the surface of that wick, where the meniscus exists, returning as vapor to the condenser.

A key problem for CPLs is vaporization within the liquid core of the evaporator. This vaporization can be caused by the back-conduction or so-called heat leak through the primary wick. The vapor can block the liquid core; this will prevent

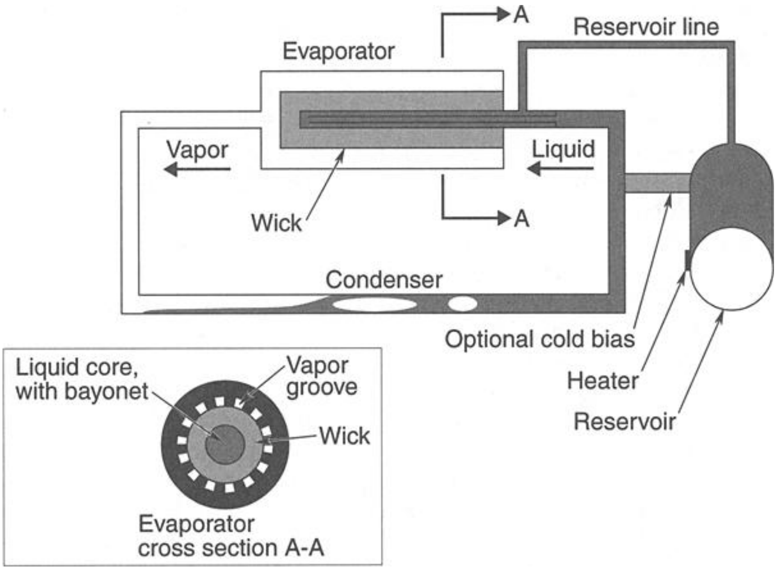

Fig. 14.12. Typical single-evaporator CPL.

proper supply of the evaporator with liquid, which in turn may result in evaporator dryout and CPL deprime (cessation of circulation).

Because this vaporization is possible, the traditional CPL employs an evaporator with a wick of polyethylene. Although the low conductivity of that material greatly reduces back-conduction, and although it is very easy to work with, its pore sizes are relatively large (15–20 μm), and consistent, inexpensive suppliers of the material are scarce. Polyethylene wicks are used in the EOS-AM, MSP, and HST missions, but future CPL wicks will probably use alternate materials such as titanium or ceramics.

CPLs feature a reservoir plumbed into the liquid line of the loop, and perhaps thermally connected to that line for temperature-control purposes, although other relatively weak sources of cold-biasing also suffice. The EOS-AM and HST CPLs, for example, use externally mounted reservoirs for mission-specific reasons, whereas in other CPL designs the reservoir is internal and cold-biased via a thermal connection to the liquid line.

In modern CPLs, the reservoir is plumbed into the evaporator itself using a very thin line, making the evaporator a "flow-through," "three-port," or "starter-pump" (a historical misnomer) design. No capillary connection links the reservoir and the evaporator, as is the case with LHPs, although additional wicks or baffles are almost always used within the CPL reservoir, to manage the two phases in that device.

The CPL reservoir is heated above the evaporator's temperature before start-up to ensure that liquid is in the wick and that bubbles in the evaporator core have been collapsed. Thus, the only void in the system exists within the reservoir before start-up. As the reservoir cools and/or the evaporator warms, boiling eventually takes place on the vapor side of the evaporator (perhaps even violently if the evaporator superheats); vapor fills the vapor line and condenser, which pushes liquid into the reservoir. Advantage is taken of this rapid liquid displacement to flush the evaporator core of bubbles that may have been created during the initial boiling or during the stressful clearing of the vapor line (the "purge surge").

The reservoir is oversized such that a void always exists within it. This makes the devices operate in a variable-conductance mode, wherein the temperature of the thermally remote reservoir controls the evaporator temperature, and the condenser floods (i.e., is thermally "blocked" with liquid, with little energy exchange occurring in the liquid region compared to the two-phase region) as needed such that the overall conductance of the device is controlled by this reservoir temperature. The EOS-AM unit is such a variable-conductance design, as were the COMET CPLs (which were never flown, because of an unrelated mission failure) and the HST design.

In the late 1980s, the fixed-conductance mode of operation was invented to deal with the problems associated with multiple CPLs operating in parallel. If too much heat is added to a variable-conductance CPL and the reservoir is oversized, then the condenser will open up too much and either lose too much subcooling or, worse, let vapor pass to the evaporator. "Fixed-conductance mode" simply means that the reservoir and charge have been sized such that the reservoir becomes hard-filled with liquid before this "overdrive" failure mode is encountered. When excessive heat is added to a CPL that operates in a fixed-conductance mode, the temperature of the loop simply rises as the reservoir ceases to control the saturation temperature of the loop. In many ways, a fixed-conductance CPL regulates its own subcooling just as an LHP does.

Unfortunately, the fixed-conductance CPL is susceptible to start-up failures when the evaporator is thermally attached to a significant mass and/or enough noncondensable gas is present. For these reasons, the fixed-conductance CPL concept was abandoned. Because LHPs can experience start-up difficulties for similar reasons, however, the lessons learned from the fixed-conductance CPL should not be lost. Fortunately, in LHPs an alternative is available that has no counterpart in CPLs—compensation-chamber (CC) cooling. Start-up of LHPs is described in more detail below.

An alternative design measure that also avoids loss of subcooling at high powers and/or warm environments was employed successfully in the HST CPL. That design relies on a control system to heat the reservoir as needed to maintain subcooling, in effect simulating the fixed-conductance mode while maintaining a void in the reservoir. The substantial length and large diameter of the liquid line in this system yielded a long time constant of approximately 20 minutes, which provided considerable latitude in the design of the reservoir heater-control system. Application of this heater-control technique to smaller CPLs remains to be demonstrated.

As with LHPs, many variations of single-evaporator CPLs are possible; this makes the technology confusing to the outsider accustomed to working within the constraints of heat pipes, which have constant cross sections and few if any plumbing or arrangement options. One would perhaps not even recognize as the same device the CPLs used on the EOS-AM, HST, and COMET missions—each was highly customized.

An even more dramatic variation was achieved in 1993 with the creation of the first cryogenic (80 K) CPL, which used nitrogen (and later was recharged and retested with neon) as a working fluid. These miniature (0.5–5.0 W) devices, which introduced new components such as the "hot" (room temperature) reservoir, are able to start with a room-temperature evaporator in an unflooded (indeed, superheated and perhaps even supercritical) loop. Flight tests of this device in 1998 proved its zero-g performance and reliable startup in microgravity.

LHP Overview

LHPs were invented in Sverdlovsk (now Ekaterinburg), Russia, by scientists from the Institute of Thermal Physics of the Ural branch of the Russian Academy of Sciences. In the early 1980s the first Russian and U.S. patents, as well as some European patents, were issued. Originally the LHP was called the antigravitational heat pipe, but in the late 1980s it was renamed loop heat pipe. A large role in LHP evolution and the adaptation of preliminary concepts to practical design applications was played by the Lavochkin Association (Khimky, Russia) of RKA (Russian Space Agency) and HPO PM Krasnoyarsk (Russia).

The traditional schematic of an LHP is presented in Fig. 14.13. A classical LHP consists of evaporator and CC assembly, condenser, and transport lines. The specific configuration of an LHP is determined by the application.

As mentioned above, both an LHP and a CPL theoretically only require wick material in the active evaporator zone; the remainder of the LHP is wickless tubing (condensers can use various designs, but they need not contain any wicks). As with a CPL's reservoir, the LHP's analogous CC normally also contains some wick structure with properties different from evaporator wick structure, but such "secondary wicks" are not strictly necessary; they merely enhance performance and robustness and help adapt an LHP to zero-g applications.

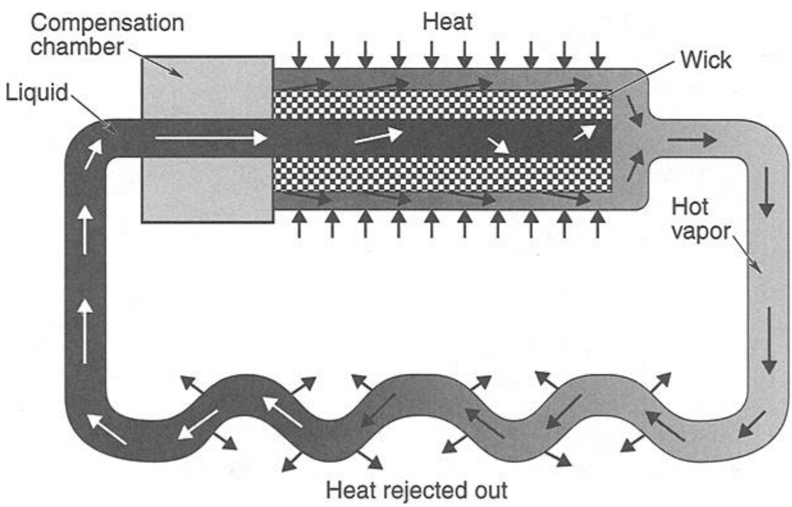

Fig. 14.13. Classical LHP with direct-condensation condenser.

Traditionally, the evaporator consists of a cylindrical metallic case with the wick inserted into it. The case then either is attached to the heat-acquisition surface or actually forms the surface itself. Several successful attempts were made to create a "flat" evaporator with platelike case; however in most cases the internal pressure of the LHP system is so high that pressure-containment considerations dominate the evaporator design, and walls of such an evaporator case would be too thick.

A network of vapor-removing channels is formed at the area of contact between body and the wick. This area is considered the active evaporator area.

The CC shares liquid with the inside of the primary wick in the evaporator (i.e., the liquid core). This sharing is accomplished either by gravitational forces or via the use of a secondary distribution wick. Vapor and liquid lines enter and exit the evaporator and CC assembly. The liquid return in an LHP flows either into or through the CC (unlike the return in a CPL, which does not flow into or through the reservoir), with intimate thermal contact with the CC's contents. The CC is a critical component: Its design has to be considered very carefully, because its sizing affects the performance of the LHP (conductance, maximum power, minimum start-up power, etc.).

The condenser of an LHP plays the same role as the condenser of a CPL: It condenses the vapor that was generated in the evaporator and transfers heat to the sink (by any means: conduction, radiation, or convection). ESA has performed a detailed study of different types of condensers for space application.[14.18] For modern space applications two types are usually considered in trade-off studies: direct and indirect. Direct condensation assumes that the condenser has been designed as tubing network (parallel or series) attached directly to the radiator facesheet (or another heat-rejecting device). Indirect condensation assumes that an additional interface is interposed between the surface of condensation and the heat sink, which in most cases is a heat exchanger to the evaporator of a heat pipe. Both approaches have advantages and disadvantages, as detailed in Ref. 14.18.

Transport lines are simply smooth tubing without capillary structure.

The selection of materials for LHP components as well as the working fluid is the subject of detailed study during the design phase. The most studied and reliable combination of LHP materials includes stainless steel, aluminum, nickel, and ammonia. This combination was experimentally proven compatible, and the compatibility minimizes noncondensable gas generation. (Noncondensable gases and physical leakage are the two most important factors that can reduce LHP lifetime.) A detailed experimental study[14.19] of LHPs with this combination of materials showed that even the most conservative predictions of the noncondensable gas volume generated at the end of life in such an LHP does not cause much distortion in LHP behavior and performance. Alternate LHP materials include nickel for the evaporator body, porous titanium for the wick, and propylene for the working fluid.

To simplify consideration of LHP operation, an LHP with a very simple, classical point design will be discussed: a single evaporator combined with CC, serial direct-condensation condenser, and semiflexible transport lines. The working fluid is ammonia.

LHP designers consider three cases of LHP operation:

- **Cold case.** In the cold case, zero power is applied to the evaporator, and condenser and transport lines are exposed to coldest environment conditions. The most conservative assumption in this case is that the entire loop (other than most of the CC) is liquid filled, including the primary wick and the evaporator's vapor exhaust grooves.

- **Hot case.** In the hot case, maximum power is applied, and the rest of the loop is exposed to the hottest environmental conditions. The assumed fluid distribution in a hot case is as follows: The vapor exhaust grooves in the primary wick, the vapor line, and the condenser are filled with vapor, while the primary wick, the liquid line, and most of the CC are filled with liquid.
- **Maximum nonoperating temperature.** In the third case, which is often a driver in LHP design, the exposure is to the maximum temperature under non-operating conditions (storage, transportation, perhaps some manufacturing process like bonding) after the loop has been charged with the working fluid. The concern with this case is that enough void must remain in the loop to avoid bursting as a result of hydrostatic pressures.

The hot case is used to size the radiator to allow rejection of the heat without overheating the payload, and without hard-filling the CC with warm (low-density) liquid (such a condition would lead to condenser blockage). In the cold case the designer must worry about the potential freezing of the system along with the requirement that some liquid must exist within the CC despite the high density of the cold fluid. The name "compensation chamber" ("hydroaccumulator" and "reservoir" are frequently used as synonyms) derives from the main purpose of that volume, to compensate for the thermal expansion of the working fluid at different operating temperatures. In other words, the main idea of the LHP is to have the CC and the fluid charge sized in a manner that provides enough liquid in the cold case to keep the evaporator wetted before start-up, yet prevent condenser blockage in the hot case.

The typical performance curve (temperature vs. power) of a classical LHP design is presented in Fig. 14.14. The shapes of the hot- and cold-case curves are

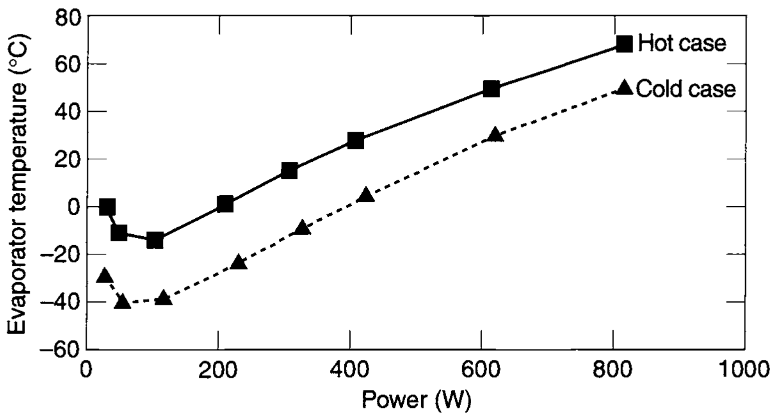

Fig. 14.14. Typical performance curves of an LHP hot case (T_{sink} = 233 K) and cold case (T_{sink} = 153 K).

identical. The only difference between the cases—operating temperature—results from different sink conditions. The performance curve of a classical LHP consists of two parts: the variable-conductance mode (the curved line at lower powers), and the constant-conductance mode (the straight line at higher powers).

The curves in Fig. 14.14 show the evaporator-case temperature as a function of heat input. This type of curve is common whenever the sink temperature is lower than the ambient temperature. In the figure, the evaporator temperature at low powers (up to about 100 W for this particular loop in this particular environment) drops with increasing power (the drop corresponds to a decrease in the overall resistance), until a minimum temperature is reached. As power continues to increase, the curve of T versus Q has a positive slope and its shape approaches a straight line; the overall resistance is nearly constant in this regime. There is a difference between evaporator and vapor temperatures as a result of the finite evaporator resistance. This difference is zero at low powers and increases linearly with power.

This particular behavior is directly related to the location of the CC and its coupling to the evaporator. When power is applied to the evaporator, a capillary pressure difference across the wick develops to sustain the pressure drop created by the hydraulic resistance of the transport lines and condenser. This capillary pressure must also sustain the gravity head of the liquid column in the return line (if the evaporator is located higher than the condenser). The pressure difference across the wick, which is the driving force of the working fluid in any LHP, also creates a corresponding temperature difference across this wick as a result of the Clausius-Clapeyron relationship. The resulting heat leak can eventually increase the CC temperature (and pressure) to the point where the driving potential of the pressure difference will not be able to move working fluid through the loop. In a real LHP system, such a heat leak is compensated by the subcooling of the liquid that enters the CC from the condenser. This subcooling is generated in the condenser by its partial "blockage." When powers applied to the LHP are low, the small flow rate in the loop means very little liquid is flowing into the CC. In such low-power cases, the CC temperature is dominated by its heat exchange with the evaporator and with the environment.

This effect can be illustrated by a simple model of the CC thermal balance. The required subcooling (Q_{subcool}) that needs to be generated in the condenser must equal the sum of the heat leaks to the CC: heat leaked through the environment (Q_{CC}), heat leaked through the wick and evaporator structure and environment (Q_{wick}), and heat applied to the liquid line ($Q_{\text{liq-line}}$):

$$Q_{\text{subcool}} \approx Q_{\text{CC}} + Q_{\text{wick}} + Q_{\text{liq-line}}. \tag{14.2}$$

The generated subcooling brought into the evaporator can be determined using the following simple equation, where m is the mass flow rate determined by the applied power and the latent heat of the working fluid ($m \approx Q_{\text{loop}}/H_{fg}$) and $\Delta T_{\text{subcool}}$ is the difference between the saturation temperature of the loop and the temperature of the liquid exiting the condenser:

$$m \cdot C_p \cdot \Delta T_{\text{subcool}} = Q_{\text{subcool}}. \tag{14.3}$$

At low powers the flow rate m is small and $\Delta T_{subcool}$ is limited by the sink temperature. Therefore the required and actual subcooling are not equal, and the thermal balance of the CC will be dominated by the parasitic heat leaks (from the wick and the environment), which will increase the system temperature to the point where $\Delta T_{subcool}$ will be high enough to satisfy the balance between the required and generated subcooling.

At zero flow there is no capillary pressure difference across the wick of a horizontal (or microgravity-based) LHP and, because saturation conditions exist on both sides of the wick, no temperature gradient. Consequently, the only heat input to the CC is from the environment, and the CC's temperature will equal the ambient temperature. As the heat is applied to the evaporator, some degree of subcooled liquid starts to enter the CC. As the heat input increases, more (and cooler) liquid from the condenser gradually lowers the temperature of the CC. The saturation temperature follows the downward trend because the subcooling production increases while the parasitic heat leaks into the CC remain approximately unchanged.

If the sink temperature is constant, the decrease in evaporator temperature is synonymous with increasing heat-pipe conductance. As with any heat pipe, the overall conductance of an LHP is determined by the evaporator and condenser conductances:

$$1/C_{overall} = 1/C_{evap} + 1/C_{con}. \tag{14.4}$$

The evaporator conductance is normally assumed to be constant; thus the increase in overall conductance with increasing power must be the result of increasing condenser conductance. This is explained by gradual displacement of liquid from the condenser (i.e., movement of the last two-phase point within the condenser) and the subsequent exposure of more condenser length for two-phase heat exchange.

At zero or very small power levels, only a short section of the condenser is active, and the remainder is filled with liquid, producing as much subcooling as possible in an attempt to compensate parasitic heat leaks into the CC, but being limited by the low flow rate.

As the power is increased, more and more condenser area becomes active and the overall conductance of the LHP continues to increase. In this mode of operation, the LHP behaves like a variable-conductance heat pipe (VCHP). The range of power over which the VCHP behavior applies and the exact nature of the accompanying temperature changes depend on the design of the LHP as well as on the current temperatures of the sink and the environment.

At a certain power the condenser is completely active, reserving only a very small percentage of its length to produce the required subcooling. Further increase in the condenser conductance is no longer possible. To reject additional power, the driving potential between condenser and sink must increase, resulting in an increased saturation temperature. From this point, the LHP behaves like a fixed-conductance heat pipe (FCHP): The temperature difference between evaporator and condenser increases linearly with power.

The physics of LHP operation were well studied in the former Soviet Union, the device's country of origin. One of the most detailed publications on LHP fundamentals is Ref 14.20.

CPLs vs. LHPs

The distinction between CPLs and LHPs is historical and controversial. Coming from different heritages, they were associated with different design philosophies. Each approach has some basic pros and cons. Because both CPLs and LHPs are relatively new to many users, difficulties associated with their application to modern spacecraft are described in this section along with their relative merits.

The approaches share many similarities;* one can even build devices that are somewhere between a traditional CPL approach and a traditional LHP approach; these have their own advantages and disadvantages. Therefore, although the following sections concentrate on the distinctions between the traditional CPL and LHP approaches from the standpoint of a potential user, remember that extensive design flexibility exists for customizing these devices, and that the distinctions are given only to help explain the fundamentals of their operation, not to replace critical consultations that should be sought when making procurement decisions.

Distinguishing the Two Traditional Loops

The basic distinction between a traditional CPL and a traditional LHP lies in the fluidic and thermal attachment of the reservoir or CC. This seemingly simple distinction has a large impact on the design and operation of the loop.

In a traditional CPL, the circulating fluid does not pass through the reservoir. Instead, the reservoir is attached to the liquid side of the loop by a small-diameter line whose time-averaged flow rate is zero under steady conditions. Although not strictly necessary, the returning subcooled liquid is usually thermally (but not fluidically) connected to the reservoir, providing a cold bias for control purposes instead of (or in addition to) any cold bias afforded by heat losses to the reservoir's thermal environment. The CPL reservoir may therefore be located anyplace. Although wicks are located within the reservoir for phase-management purposes, no wick connections are between the reservoir and the evaporator, much less any other part of the loop. The loop is sized such that liquid with adequate subcooling is always supplied to the evaporator, perhaps via the use of a feedback-controlled heater on the reservoir, as with the HST CPL.

In a traditional LHP, on the other hand, the CC (the LHP equivalent of the CPL reservoir) must have a good flow path between the liquid within it and the main evaporator wick to provide reliable operation in transient modes. Unless gravity is available to maintain this path, a secondary wick is used. Because the liquid return flow often flows into or through the CC, the thermal connection between the liquid line and the CC can be essentially infinite. The evaporator liquid core is normally considered to be part of the CC. The loop is sized and charged with working fluid such that the CC can never be completely filled with liquid nor completely void of liquid.

Start-Up

In addition to advantages that CPLs and LHPs offer over mechanically pumped systems and conventional heat pipes, these systems introduce a particularly

*For example, a fixed-conductance CPL with a void in the evaporator core is thermodynamically identical to an LHP. Although this is an "off design" condition for a CPL, it is a normal operating point for an LHP.

important feature: start-up assurance. Two-phase systems require some degree of preconditioning to achieve a proper and reliable start-up.

From the very early days of two-phase loop development, an understanding of the start-up phenomena was the focus of technology developers on both sides of the Atlantic. This attention was warranted because the ideal LHP starts without any concern for orientation, history, or preconditioning. An LHP does not require such extensive preconditioning and in most cases does not require any at all. As soon as the temperature gradient between the CC and the evaporator (i.e., the temperature gradient across the wick) is big enough to build up the pressure difference required to initiate circulation, the LHP starts. The only problem with LHP start-up is achieving this threshold gradient. CPLs, on the other hand, cannot start without intentional preconditioning of the system. This preconditioning consists of flooding the capillary pump and the vapor line with liquid by heating the reservoir a few degrees (perhaps 5 to 15°C) above the evaporator for a period lasting from 30 minutes to 3 hours.[14.21–14.25]

The start-up process for two-phase systems is very complicated: Many random and design-specific factors can prevent successful start-up. Both CPLs and LHPs can experience problems starting with very low powers (heat flux is a very important parameter), or in the presence of heavy masses attached to the evaporator (the thermal mass affects the evaporator heating rate, and the rate at which heat penetrates into the CC). However, the LHP is especially susceptible to these problems when the low-power start-up is experienced following diode operation or very cold conditions, when large amounts of gas are present, or when the evaporator is elevated above the condenser.

A successful start-up is characterized by the generation and maintenance of an adequate temperature difference across the evaporator. This threshold temperature difference is evaporator design-specific and is also affected by application requirements, environmental conditions, and even prior usage history. Because of the unknown state of the wick core, start-up can occur in a number of ways, and the exact scenario is unpredictable without knowledge of the recent history of the loop. (This does not mean that computational predictions cannot be used as a design tool, only that conservatisms and enveloping are required to accommodate uncertainties.) Some LHP designs, especially those operating at mid and high powers, do not require any additional start-up precautions.

One of the first publications that summarized the basic understanding of the processes that occur in the LHP evaporator before, during, and after the start-up event was the referenced paper by Maidanik *et al.*[14.26] Maidanik *et al.* presented another paper on this subject at the following ICES conference.[14.27] These two publications were the first to describe the basic physics of the LHP start-up process. Subsequently, a number of studies were performed to investigate the transient behavior of LHPs and to create a better understanding of the start-up phenomena.

As mentioned above, LHPs were always intended as devices that would neither cause start-up problems nor require special treatment. They were therefore initially represented as self-starting, worry-free heat-transfer devices that (when compared to CPLs) do not require complicated start-up procedures. This declaration was repeated in presentations and papers for several years: "An LHP does not require any in-orbit operation approach because it is a passive, self-starting device that does not require external power or any preconditioning. An LHP starts when the heat load is applied and operates until the heat load is removed or the sink conditions cannot maintain the operating temperature range." This statement contrasted strongly with descriptions of CPLs, some of which at the time were suffering from start-up difficulties despite the care given to preconditioning.

Unfortunately, recent investigations have demonstrated that LHP start-up is not so simple, and its requirements must be considered more carefully. In recent years, basic understanding of the start-up behavior was further developed. This research has provided LHP designers and users with more confidence, trust, and flexibility in utilizing LHPs in realistic applications.

In a nutshell, start-up is not to be treated lightly for either device: Both can exhibit anomalous behavior, as is described below along with possible solutions. One of the key distinctions of the LHP in this regard, however, is the availability of certain success-enhancing design measures not applicable to CPLs, giving LHPs an edge.

The dynamics and design solutions associated with LHP start-up are described in more detail below. Start-up is a greater concern with LHPs than with CPLs, but although it is an area of concern for both devices, it is more frequently overlooked by engineers considering LHP solutions. It is important to note that many of the same physical processes that occur in LHPs during start-up occur as well in CPLs.

Assisted Startup: Active Design Measures

The wide range of LHP and CPL applications includes design cases in which these loops may require start-up assistance. The assistance may be necessary because of an insufficient level of evaporator heat flux, an excessively high "heat leak" through the wick into the liquid core or, in case of an LHP, an unfavorably warm CC (the liquid core of the evaporator is usually considered to be an integral part of the CC). The heat flux applied to the evaporator should be sufficient to generate the required superheat on the outer surface of the primary wick. However, in some cases, the heat flux applied is insufficient to superheat the liquid on the outer surface of the evaporator or to overcome gas or adverse tilt without also heating the liquid core of the evaporator. This lack of sufficient heat flux can result from insufficient power being applied to an evaporator with a large evaporation surface, or to a large mass attached to the evaporator, or to an attached redundant system (i.e., another LHP, CPL, or other heat-transfer device attached to the same heat source for redundancy).

Certain combinations of initial conditions and mission scenarios can be dangerous because they can, in some cases, lead to start-up problems. Fortunately, these potential problems are related to the difficulties associated with creating the minimum required temperature gradient across the wick, so active design measures can be implemented to assist the two-phase system in establishing this gradient—either by heating the outside of the wick or by cooling the inside of the wick (effectively, by cooling the CC).

The first method, heating the outside of the wick, is the simplest and most popular. A starter heater is mounted onto the evaporator surface. This starter heater need not be of high wattage; the main requirement for this heater is to add a concentrated local heat flux to the normal distributed heat load on the evaporator, at least when superheat for nucleation is the concern. This concentrated flux heats the local area under the heater much faster than it creates heat leaks into the wick. As a result, local superheat is created, and as soon as it exceeds the critical value, the vaporization will start. The initial vapor bubble will grow explosively if the liquid has been superheated, expanding into the rest of the vapor space of the evaporator. In an LHP or in a three-port CPL evaporator system, this expansion and extensive vaporization will force the colder liquid from the condenser and liquid line to enter the evaporator core (and in the case of an LHP, the CC as well), initiating loop circulation.

Starter heaters for specific applications can have different designs, but the main requirement is to maximize the heat flux over a localized area. A number of different types of starter heaters, from a bulky cartridge resistance heater to a very light but fragile ceramic heater, were proven to be sufficient aid for various designs. Normally, film heaters cannot provide the desired heat flux and can be used only with some design restrictions. Starter heaters can be defeated by thick, highly conductive evaporator cases and by tight thermal connections to massive plates and payloads (or even to redundant LHPs and traditional heat pipes), so careful design is required.

Unfortunately, because they must necessarily be small, starter heaters (though they may be effective at overcoming incipient superheat thresholds) are less effective at establishing gradients across the wick as needed to overcome gas and adverse tilt conditions. In those cases, the alternate strategy may be applicable: cooling the CC. This second design measure is neither as simple nor as common as the first and can only be applied to LHPs. In the early 1990s this method was proposed by Russian authors to enhance the start-up performance of LHPs designed for applications with special requirements.[14.28]

The use of thermoelectric coolers (TECs, also called Peltier coolers) to cool the CC became well known. This method was even "baselined" by a few companies as a solution for their LHP start-up problems. The cold side of the thermoelectric element is attached to the CC surface, and the TEC's hot side is attached to the evaporator via a thermal shunt. The electric energy applied to the TEC will not only cool the CC but will heat the evaporator as well, enhancing the positive effects of CC cooling.

This active design measure has two disadvantages. The major disadvantage is the need for a special high-current, low-voltage power supply. The other one can be major without proper design: The TEC can become a very effective condenser for vapor generated on the internal surface of the primary wick. If a vapor bubble is present in the wick core, the TEC will assist an internal "heat-pipe effect" that can hinder the creation of the required temperature difference. If established, this internal heat-pipe effect eliminates all advantages of the TEC approach because it extracts heat from the evaporator.

Subcooling and Overall Conductance

CPLs and LHPs require different degrees of subcooling, because of their different thermodynamic processes. An LHP requires some subcooling to compensate the heat leak from the evaporator and environment to the CC, which is usually very small (less than a watt). However, because flow rates are so low and so little energy is available via the sensible cooling that subcooling represents, these fractions of a watt can translate into several degrees of required subcooling. This same sensitivity applies to CPLs, although their requirements for subcooling are even larger.

Theoretically, a CPL should require no subcooling, and because it normally has no void in the liquid side of the wick, it can even operate with superheated liquid (this has, in fact, been demonstrated). However, in practice a CPL requires subcooling both (like an LHP) to compensate for the heat leaks into the liquid part of the system, including back-conduction through the wick, and (unlike an LHP) to guarantee that any vapor bubbles that occasionally appear in the liquid core will be collapsed or at least will not grow without bound. In other words, under ideal conditions a CPL requires less subcooling than an LHP. However, in practice a CPL's sensitivity to vapor blockage means that, to operate robustly, more subcooling must be designed into the system than is theoretically required. Given the same wicks, a CPL will require more subcooling and therefore have a lower overall loop conductance. (However, recall that CPLs traditionally have been made with lower conductivity and lower-pore-size plastic wicks, so the above generalization has not always been evident.)

To appreciate basic differences between a CPL and an LHP, consider the addition of a small amount of heat to the liquid line in both systems. The extra heat appears to have no effect on a CPL; temperatures do not change. This makes the performance of CPLs relatively easy to predict. In essence, some of the "overdesign" in subcooling is used to compensate for this heat addition. However, if the amount of heat is raised sufficiently (on the order of perhaps 10 W), the CPL will deprime (i.e., fluid will cease to circulate), and the evaporator temperature will increase without bound unless a reprime cycle (heating of the reservoir) is initiated. In other words, once the liquid-line heating either generates bubbles in the liquid line or causes a preexisting bubble in the evaporator core to grow excessively, the evaporator wick will be starved of liquid and will deprime.

An LHP acts very differently. Even small amounts of energy (e.g., tenths of a watt) added to the liquid line cause a noticeable change in temperatures and overall conductance, especially at low powers. However, continuing to add energy does not result in an abrupt deprime of the LHP as it does the CPL: The temperature of the evaporator will simply continue to grow relative to the condenser. In effect, the LHP autonomously changes its state as needed to generate enough subcooling to counterbalance the applied heating, a feat that cannot be matched by a CPL except by addition of a reservoir heater controller. If a large enough heat rate is applied to the liquid line, however, an LHP will eventually deprime, but it will be a more graceful shutdown than would be the case with a CPL.

The performance of a CPL is like a step function: Either it continues with no change, or it fails. An LHP's performance is more like a ramp function: It changes in response to environmental (thermal or gravitational) changes, but it does not so much fail as degrade gracefully in response to adverse conditions.

Robustness

The LHP is arguably the more robust device of the two. If a CPL does not have adequate subcooling on average, perhaps because of a design or implementation problem, it will deprime. On the other hand, if an LHP does not have adequate subcooling, its operating temperature will simply rise to create sufficient subcooling. To oversimplify, minimum subcooling graphically represents a "cliff" to a CPL, a point beyond which the device cannot operate, whereas an LHP (in its so-called "autoregulating" mode) will always adjust itself to maintain the required subcooling.

Ultimately, the CPL is sensitive to bubbles within the evaporator core, lacking a good mechanism for removing or collapsing them other than subcooling. The LHP responds more gracefully to bubbles. The loop conductance decreases, but does not abruptly shut down.

Reservoir Location

A disadvantage of the LHP as compared to the CPL is that the CC of an LHP must be physically very close to the evaporator because they share fluid either directly or via a relatively weak capillary link. This requirement can cause integration difficulties, because the CC is often relatively large. Worse, it is sensitive to heat gains: Packaging the evaporator and CC assembly next to a heat source can be difficult in some applications, because real estate near the source is often scarce. For example, embedding the evaporator and CC assembly within honeycomb panels is difficult because a strong thermal connection must be made with the evaporator, while the CC must be thermally isolated from the source.

The CPL's reservoir, on the other hand, is connected to the evaporator via a very-small-diameter line that can be arbitrarily long. Also, the reservoir in a CPL is not as sensitive to heat leaks. The evaporator is a separate unit that is amenable to integration into planar baseplates. Therefore, in some applications CPLs can be easier to integrate than LHPs.

Controllability

Both CPLs and LHPs can be controlled to maintain set-point temperatures. Because subcooled liquid flows directly through its CC, an LHP usually requires somewhat greater heater power to maintain a set point in cold modes of operation,* but in both cases only a very small fraction of the loop power is needed to control the set point. The ease with which such control can be added to these loops may confuse engineers familiar with traditional heat pipes. In those devices, temperature control is achieved not simply by adding a thermostatic heater, but by effecting a complete change in design (from fixed to variable conductance) with a corresponding jump in cost and complexity. With CPLs and LHPs, control is rather easily added to the design if required (Fig. 14.15).

It is useful to note differences in the process by which control is achieved. The heating of the CPL reservoir causes the additional thermal blockage of the condenser, and accordingly affects the maintenance of the system temperature level. (That level changes as a result of the reduction of the effective condensation area.) The amount of heat required on the reservoir to maintain a set point is therefore a

*On the other hand, in an LHP the heat for such control can be extracted in part from the payload dissipation itself by a thermal connection between the vapor and liquid lines, or even via the introduction of a three-way valve.

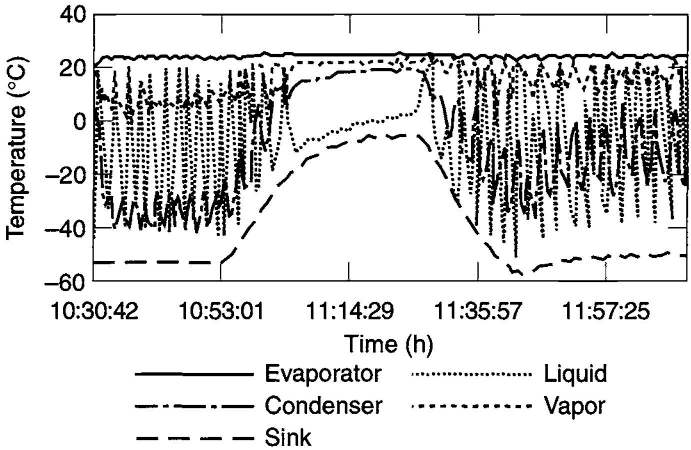

Fig. 14.15. Typical LHP behavior in case of active regulation (maintain the evaporator set point, 21°C, at 100 W and sink transient –60°C to +5°C to –60°C).

function of its local thermal environment and is unrelated to the loop operation. The heating of the LHP CC also causes condenser blockage but by a somewhat different mechanism: the need to generate sufficient subcooling to offset the heat added to the CC. This statement is illustrated in Fig. 14.16.

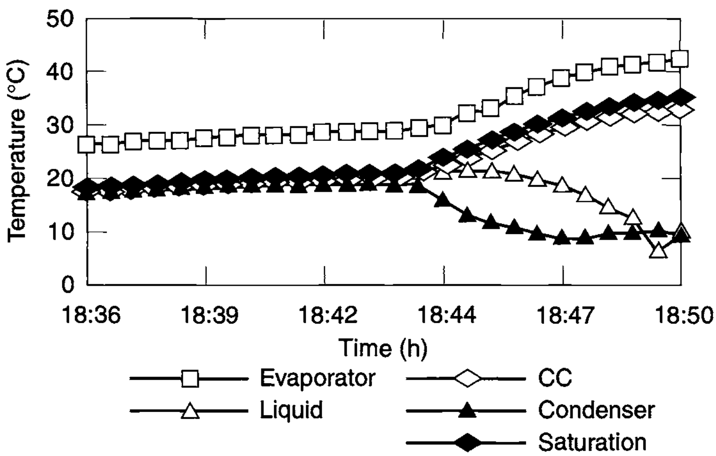

Fig. 14.16. Typical LHP temperature history with the heating of the CC, 30 W, applied (the LHP was running at 400 W).

Selecting a Design

The thermal engineer is responsible for selecting the simplest and safest system that will suffice; therefore, simple fixed-conductance heat pipes should be chosen if possible. However, given the occasional disadvantages of heat-pipe systems (many of which are apparent only upon producing a complete design from scratch that takes advantage of the unique features of LHPs and CPLs), a loop design may be compelling if not enabling.

The top-level selection of an LHP or CPL architecture should be made with care. Remember that certain mission-specific requirements may dictate the choice, and that these devices both have tremendous design flexibility to accommodate specific requirements. This flexibility includes the creation of devices that are not easily classified but exhibit both CPL and LHP characteristics. At the risk of overgeneralizing, however, the following ideas should serve as guidelines.

LHPs are simpler and usually more robust than CPLs. They exhibit turnkey start-up, tolerance of inadvertent liquid-line heat leaks or loss of condensation, etc. Somewhat like a thermal analogy to electrical extension cords, they are excellent choices for retrofits or cases where any improvement is helpful though the path between the source and the sink may be tortuous. Just as simple heat pipes should be attempted before using loop designs, LHPs should be attempted before using CPLs.

CPLs find application in cases where more mission involvement is either tolerated or desired, such as missions demanding active reprime contingencies, and of course missions that are tolerant of preconditioning requirements before the initial start-up. The remoteness of a CPL's reservoir may cause start-up disadvantages but can also be the source of some of the CPL's advantages. The reservoir of a CPL can be attached remotely in almost any convenient location and can thus enable the design to avoid LHP difficulties caused by the need to locate a CC (that is rather sensitive to heat additions) with an evaporator in areas where real estate is usually limited.

For example, in the case of a start-up cryogenic loop, the ability to collocate the reservoir near the condenser instead of the evaporator (thus avoiding the need for a separate cryocooler to precool the CC of a cryogenic LHP*) provides CPLs with a design advantage at those temperature ranges.

Analysis Tools

The original developers of LHP technology, working at the Russian company TAIS, created a single-evaporator LHP model using a Pascal interpreter more than 10 years ago. Since then, this model has been upgraded several times. Swales Aerospace developed several versions of a mathematical model in spreadsheet format. Other companies develop their own software in-house.

C&R makes available a free SinapsPlus "prebuilt" model with graphical spreadsheet-like access to an underlying SINDA/FLUINT thermal/ hydraulic model of a generic LHP that allows users to estimate the performance of an LHP in

*A competing LHP concept is to use two or more loops in parallel with different working fluids: One (perhaps using propylene) is used at high temperatures to bring the temperature of the other loop's CC down to where it can begin to operate.

steady-state and transient modes. This prebuilt is intended both as a demonstration of LHP modeling techniques and as template for trade-study and system-integration models.* It can be used with or without SinapsPlus, and requires no software purchases.

These models take into account both the energy balance of the CC and the pressure drop along the fluid paths in the loop. It is impossible to achieve good analytical correlation of the predicted thermal performance of an LHP to test data without tracking minor heat leaks and boundary conditions (especially of the liquid line and the CC) and accurately modeling the pressure drops in the system, which affect the back-conduction term in the CC. Short time-scale thermal/hydraulic transients (such as start-up phenomena, noncondensable gas effects, condenser quenching, etc.) have been analyzed using SINDA/FLUINT (Refs. 14.29 and 14.30, for example), but often this level of detail is needed by CPL/LHP developers and researchers, not users.

SINDA/FLUINT is similarly used for steady-state and transient CPL analyses, although no starting-point CPL prebuilt is yet available, in part because much more variation is available in CPL designs. Models built for various publicly funded systems (EOS-AM, CAPL II, CAPL III, HST) are available, however. C&R provides free training notes for use in modeling capillary systems such as LHPs and CPLs using SINDA/FLUINT.

In modeling CPLs, one need not track pressure losses and heat leaks to predict thermal performance, provided that adequate margin is applied to the design in the first place. However, some analysis cases for CPLs are not normally applied to LHPs. These include start-up from a hard-filled state (a case that is often adequately analyzed using pseudosteady bracketing studies) and pressure oscillations. (The latter phenomena may also occur in an LHP, but they are usually less of a concern.)

In both CPLs and LHPs, rather simple models suffice to characterize the transport lines, evaporators, and reservoirs for steady states or thermally dominated transients. More detail is often needed, however, in modeling condensers, where thermal performance is a function of layout (which can be quite varied and customized), bonding and integration options, orbital environment, etc. Accurate predictions of subcooling production are important for both systems, but especially for LHPs. Fortunately, the challenges associated with detailed condenser modeling are more related to the size of the model than to the complexity of concepts and modeling options. For such detailed condenser/radiator analyses, it is convenient to use a traditional thermal tool such as SINDA combined with the 1D line layout options in Thermal Desktop, with FloCAD. Figure 14.17 depicts an LHP with a 1D serpentine condenser on a 2D discretized plate (finite-difference in this case, although finite elements may also be used). Parallel, manifolded condensers may also be used, along with flow-control devices that assure even distributions in parallel-condenser designs.

More detailed information on simulation is provided in Ref. 14.31.

*For example, it was used as a template by Aerospatiale to create a model of the STENTOR multi-evaporator LHP.

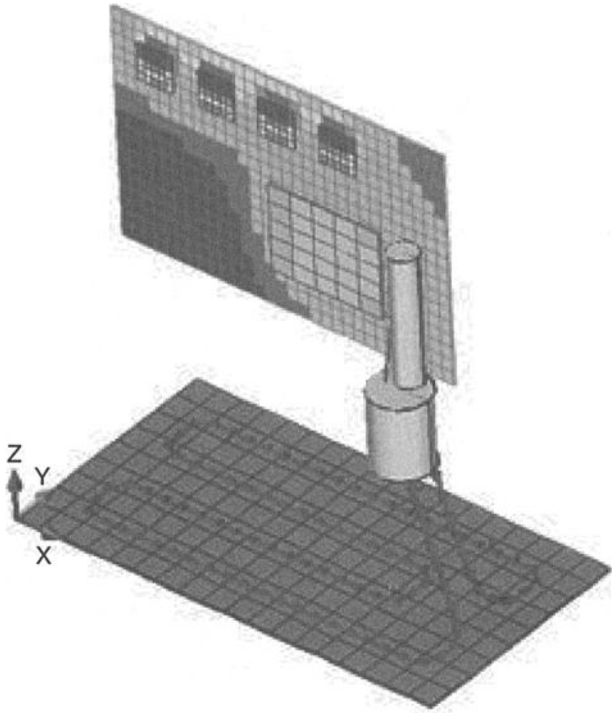

Fig. 14.17. Geometric (CAD-based) LHP thermal/fluid model, including serpentine condenser.

Conclusions

As a result of their growing acceptance, CPLs and LHPs continue to be active subjects of research and development. Current areas of research include:

- reliable lightweight low-pressure-drop parallel condensers
- freeze-tolerant radiators
- advanced high-performance, low-conductivity wicks
- evaporators of different aspect ratios (long, thin, flat, miniaturized, etc.)
- heat-transfer performance improvements
- miniature systems
- simplification of the technology, for adaptation to mass production
- multi-evaporator and multicondenser systems, including reversible loops and large-scale isothermalizers
- cryogenic systems
- high-temperature systems

CPLs and LHPs offer tremendous advantages to many thermal-control applications, especially when trained designers consider these loop transport systems early in the design process, and not just as replacements for traditional heat-pipe technology.

CPLs and LHPs are very similar devices, and they in fact represent two extremes in a spectrum of design possibilities. However, each device type has specific advantages and disadvantages, and both will continue to find application in future missions because of the broad range of mission requirements and constraints.

References

14.1. A. Abhat, M. Groll, and M. Hage, "Investigation of Bubble Formation in Arteries of Gas-Controlled Heat Pipes," AIAA Paper 75-655, May 1975.

14.2. E. W. Saaski, "Arterial Gas Occlusions in Operating Heat Pipes," AIAA Paper 75-657, May 1975.

14.3. J. E. Eninger, "Menisci Coalescence as a Mechanism for Venting Noncondensable Gas from Heat-Pipe Arteries," AIAA Paper 74-748, July 1974.

14.4. P. Brennan and E. J. Kroliczek, *Heat Pipe Design Handbook* (B&K Engineering, Towson, MD, 1979).

14.5. F. Edelstein, "Heat Pipe Manufacturing Study," Grumman Aerospace Corp., Final Report to NASA, NAS5-23156, August 1974.

14.6. S. W. Chi, *Heat Pipe Theory and Practice: A Source Book* (Hemisphere Publishing Co., McGraw-Hill, New York, 1976).

14.7. P. Dunn and D. A. Reay, *Heat Pipes*, 2nd ed. (Pergamon Press, Oxford, United Kingdom, 1978).

14.8. R. S. Gaugler, U.S. Patent Application (21 December 1942). Published U.S. Patent No. 2,350,348 (6 June 6 1944).

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

14.10. T. P. Cotter, "Theory of Heat Pipes," Los Alamos Scientific Laboratory Report LA-3246-MS, February 1965.

14.11. B. D. Marcus, "Theory and Design of Variable Conductance Heat Pipes," TRW, NASA Contractor Report CR-2018, April 1972.

14.12. A. Basiulis, "Unidirectional Heat Pipes to Control TWT Temperature in Synchronous Orbit," *Symposium on Thermodynamics and Thermophysics of Space Flight*, Palo Alto, March 1970.

14.13. B. Swerdling, R. Kosson, M. Urkowitz, and J. Kirkpatrick, "Development of a Thermal Diode Heat Pipe for the Advanced Thermal Control Flight Experiment (ATFE)," AIAA Paper 72-260, April 1972.

14.14. P. J. Brennan, E. J. Kroliczek, H. Jen, and R. MacIntosh, "Axially Grooved Heat Pipes—1976," AIAA Paper 77-747, June 1977.

14.15. J. Alario, R. Haslett, and R. Kosson, "The Monogroove High Performance Heat Pipe," AIAA Paper 81-1156, June 1981.

14.16. E. J. Kroliczek, J. Ku, and S. Ollendorf, "Design, Development, and Test of a Capillary Pump Loop Heat Pipe," AIAA Paper 84-1720, June 1984.

14.17. G. P. Peterson, "Analytical Comparison of Three External Artery Heat Pipes," AIAA Paper 87-1616, June 1987.

14.18. A. A. Delil *et al.*, "High Efficiency, Low Pressure Drop Two-Phase Condenser," Final Report NLR CR 96002 L, 1986.

14.19. W. Bienert *et al.*, "Experimental Study of Loop Heat Pipe Behavior in the Presence of Noncondensible Gas," *10th International Heat Pipe Conference*, Stuttgart, Germany, 1997.

14.20. Y. Maidanik *et al.*, "Theoretical Basis and Classification of Loop Heat Pipes and Capillary Pumped Loops," *10th International Heat Pipe Conference*, Stuttgart, Germany, 1997.

14.21. EOS CPHTS Reliability Analysis, Lockheed Martin, 1999.

14.22. Reliability Analysis, EOS-AM Spacecraft Thermal Control Subsystem, Lockheed Martin Astro Space, Design Note EOS-DN-REL-15A, Rev. A, 1 November 1995.

14.23. Failure Modes and Effects Analysis (FMEA), EOS-AM Spacecraft Thermal Control Subsystem, Lockheed Martin Astro Space, Design Note EOS-DN-REL-16A, Rev. A, 1 November 1995.

14.24. W. J. Krotiuk and J. E. Fredley, "EOS-AM Spacecraft Capillary Pumped Loops," presented at *Spacecraft Thermal Technology Workshop, The Aerospace Corporation*, March 1996.

14.25. J. E. Fredley and C. E. Braun, "A Low Pressure Drop Heat Exchanger with Integral Heat Pipe," *1988 ASME National Heat Transfer Conference*, Houston Texas, 24–27 July 1988.

14.26. Y. Maidanik, Y. Fershtater, V. Pastukhov, and M. Chernysheva, "Experimental and Theoretical Investigation of Startup Regimes of Two-Phase Capillary Pumped Loops," Paper 932305, *23rd International Conference on Environmental Systems, Society of Automotive Engineers*, 12–15 July 1993.

14.27. Y. Maidanik, N. Solodovnik, and Y. Fershtater, "Investigation of Dynamic and Stationary Characteristics of a Loop Heat Pipe," *ICES Conference*, 1994.

14.28. E. Y. Kotlyarov and G. P. Serov, "Methods of Increase of the Evaporators Reliability for Loop Heat Pipes and Capillary Pumped Loops," Paper 941578, *24th International Conference on Environmental Systems, Society of Automotive Engineers*, June 1994.

14.29. J. Baumann *et al.*, "Noncondensible Gas, Mass, and Adverse Tilt Effects on the Start-up of Loop Heat Pipes," SAE 1999-01-2048.

14.30. J. Baumann *et al.*, "An Analytical Methodology for Evaluating Start-Up of Loop Heat Pipes," AIAA 2000-2285.

14.31. J. Baumann *et al.*, "Steady State and Transient Loop Heat Pipe Modeling," SAE 2000-ICES-105, July 2000.