Passive Oscillatory Heat Transport Systems

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Abstract. An underdeveloped class of oscillatory passive heat transport cycles are discussed that have the potential to transport significantly higher heat loads than current heat pipes. Prototype cycles employing inferior working fluids have demonstrated transport of higher heat loads over significantly greater distances than similarly sized heat pipes (including CPLs and LHPs) employing ammonia. Most of the proposed cycles do not require capillary forces to circulate the working fluid. They are also relatively insensitive to gravity and might best be compared to thermal systems using mechanical pumps. The history of development to date of such cycles is presented in relation to other approaches under consideration for various semi/passive thermal control applications. Specific operational characteristics of a select loop are presented. The obvious pros and cons of these systems are discussed as well as potential applications—particularly as regards electronics cooling.

REVIEW

The extrapolated challenges for advanced electronics thermal control have been well articulated (Mudawar 2000). As the spectrum of cooling requirements continues to widen, designers are forced to remain receptive to new approaches with unique or niche performance advantages over existing and/or competing systems. From small- to large-scale systems, traditional heat pipes (including CPLs and LHPs), thermosyphons, and mechanically pumped loops represent the majority share of the phase-change thermal control/transport solutions.

Traditionally, for relatively small heat loads < O(10kW) and transport distances, heat pipes have proven excellent options for simple, reliable, and quiet thermal energy transport and temperature control at low to moderate heat fluxes. The common limitation for these devices is a capillary pressure limit, which continues to be extended through the development of higher performance wick materials and geometries. Of course, the capillary driving force is precisely the advantage of the traditional heat pipe in that no pump power and no moving parts are required for the essentially silent operation routinely achieved. High heat loads may be transported using heat pipe networks and large high power designs are being considered (Ottenstein, 2000). Thermosyphons, like heat pipes, are excellent passive cycles and can be applied at small and large scales, but at typically low heat fluxes. Thermosyphons exploit buoyancy-driven convection made possible by an acceleration field (i.e. gravitational, centrifugal, etc.). Thus it is the acceleration field strength and orientation that limits the performance of such systems, for example, preventing them from being able to transport heat 'against gravity' on Earth, or completely preventing their operation onboard low-gravity spacecraft.

For high heat loads ~O(MW) mechanically pumped loops are arguably unchallenged. Despite a pump power penalty and the 'blemish' of moving parts, mechanically pumped loops have demonstrated their value in thermal control systems for over 100 years. It is traditionally undisputed that mechanically pumped loops are the method of choice for large heat loads transported over long distances. Even at 'low' total heat loads, mechanically pumped cycles may be designed to transfer heat at incredibly high heat fluxes (Mudawar, 2000) by exploiting forced convection. Microscale pumps have been designed and are being developed rapidly. Regular improvements in pump design continue to decrease pump wear and increase pump life, and unique pump designs, particularly on the microscale, indeed possess a 'minimum' of moving parts (Forster, 1999).

A POTENTIAL NICHE FOR OSCILLATORY SYSTEMS

There are numerous metrics employed to rate the performance of thermal solutions, to which one must add economic considerations. Such metrics are usually only of value when restricted to a specific cooling application with specified requirements, i.e. terrestrial silicon-based electronics cooling, reactor thermal control systems (TCS), solar heating, etc. It is becoming increasingly difficult to make a quick and best selection of a cooling or heating cycle due to the increasing variety of approaches available and the fact that, in general, traditionally large scale mechanically pumped cycles are being miniaturized while the traditionally small scale passive cycles are being scaled up. Due also to the increasing criticality of the thermal design, it is natural now to categorize systems by multiple criteria such as total heat load *and* expected heat flux at the source/sink. In doing so one might imagine a regime map of heat pipe and mechanically pumped loop thermal solutions as shown in FIGURE 1. In this conceptual map the upper limits of heat flux are set by physical performance limitations such as pool boiling heat flux for heat pipes and forced convection phase change heat flux for mechanically pumped loops, which can differ by more than two orders of magnitude. The lower limits for the different regimes might be set by economics. For example, a mechanically pumped cycle may not be cost competitive to a heat pipe at intermediate-to-low powers and fluxes. Similarly, a heat pipe will not compete against a passive solid conductor at low powers and fluxes. It may also be impractical for a heat pipe system to transport high heat loads (right vertical limit, FIGURE 1). For moderate-to-low powers at moderate-to-high heat fluxes it is possible to envision a regime in which heat pipes are incapable and mechanically pumped cycles are uneconomical—region A, FIGURE 1. (Similar arguments may be constructed comparing transport length versus total heat rate, or pitting system mass, volume, reliability, etc. against both heat flux and/or total power, etc.)

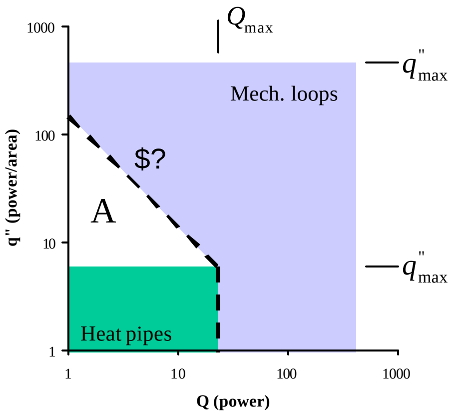


FIGURE 1. Conceptual TCS Regime Map.

It appears to be this regime of low-to-moderate heat rate, moderate-to-high heat flux, and moderate-to-high transport distance that several novel oscillatory thermal device concepts might grow to occupy. In this paper a selection of oscillatory thermal systems will be briefly reviewed. At smaller scales such devices are sometimes called 'nontraditional heat pipes' and have been referred to as oscillating or pulsating heat pipes in the limited literature discussing them. However, because such loops can be enormous, the term heat pipe seems inappropriate and the use of 'loop,' 'cycle,' and even 'system' is more fitting and more frequently used. An early distinction is made between oscillatory loops that are in part capillary controlled and those that are not. The discussion then focuses on the latter where characteristic performance data for a prototype loop is provided to support the obvious attractive features of an oscillatory approach. The potential for a niche for oscillatory cycles is then discussed followed by a list of obvious limitations and concerns.

REVIEW OF SELECT OSCILLATORY CYCLES

Oscillartory means for pumping fluids date back at least as far as 1698, with Savery's fire engine devised to pump water against gravity (Balmer, 1990). Since then, a fair number of techniques to exploit a temperature difference to circulate a working fluid have been conceived and demonstrated. The class of devices discussed here are those that develop a pressure difference in the system resulting from the condensation and/or evaporation of a working fluid in sealed or otherwise confined chambers. Such systems may be designed to be passive, but they in general do not rely on capillary forces, but rather the approximate magnitude of $\Delta P/\Delta T|_{sat}$ over the available temperature difference.

Of the considerable number of oscillatory heat transport systems in the patent literature only a few will be discussed to represent the types of systems that might find application to electronics cooling. Oscillatory thermal systems that will not be addressed herein are those seemingly intended for use on the large scale such as that of Garriss and Garriss (1978), systems devised primary for doing work on the working fluid as opposed to transferring heat as in the push pump of Jaster et al. (1974), and other systems that may require an oscillatory heat source as that of the push pump of Benner and Martin (2000).

T-System

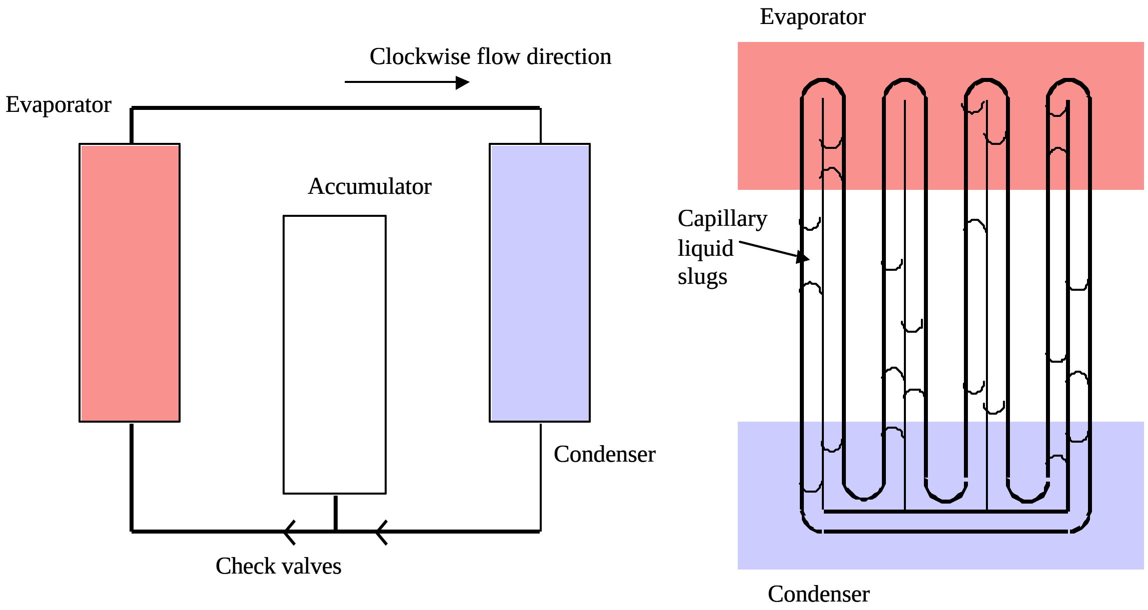
The "T-System" was proposed, demonstrated, and published by Tamburini (1978) as a novel passive means to transport heat. It is of historical interest that the scheme was first envisioned in part for cooling applications aboard spacecraft. A simplified schematic of the loop is shown in FIGURE 2. The schematic, like those to follow, does not show the preferred configuration of the loop for heat pipe applications such as electronics cooling, but rather a configuration from which it is easiest to describe the principle of operation.

The loop consists of an evaporator, condenser, and two-phase condensate accumulator. Check valves upstream and downstream of the accumulator permit flow only in the clockwise direction as sketched. With steady heat input to the evaporator the volatile working fluid is vaporized under nonequilibrium conditions leading to a local rise in pressure. Because the vapor production rate in the evaporator temporarily exceeds the condensation rate in the condenser, high pressure vapor flowing into the condenser condenses, but also forces condensate through the upstream check valve and into the accumulator. As the liquid level in the evaporator decreases, vapor generation in the evaporator decreases. Thus, the condenser condensation rate temporarily exceeds the vapor production rate causing a nonequilibrium decrease in evaporator/condenser pressure. This decrease in pressure of the evaporator/condenser leads to the injection of liquid from the now 'high pressure' accumulator through the downstream check valve and into the evaporator. This process is repeated on a periodic basis despite uniform steady heat input. Temperature and pressure fluctuations accompany each cycle. The T-System is self-starting like a heat pipe, relatively insensitive to liquid subcooling, does not require capillary dimensions, and requires only 2 check valves for operation. To the knowledge of the author, the limits of peak heat rate, flow resistance, and transport distance have not been quantitatively established.

Pulsating Heat Pipe

Another, perhaps more elegant, oscillatory device was introduced by Akachi (1990) and improved by Akachi (1993), and is rightly called a pulsating heat pipe. Akachi's loop is represented schematically in FIGURE 2 where the conduit serpentines into and out of the heat absorbing (evaporator) and heat rejecting (condenser) zones. The loop is a passive device with no mechanical parts, though any number of check valves or other passive valves may be employed at various locations along the tube. Heat transport across the device is complex and chaotic. With the addition of heat along the evaporator section, liquid in the tubes there superheats and explosively evaporates, forcing the still evaporating slug towards the condenser like a shrinking piston which in turn propels condensed liquid in an adjacent leg of the condenser back towards the evaporator where the process is repeated. The transport phenomena is indeed complex, combining multiple instabilities and vapor recoil forces and would be an acedemite's dream if not for the potential applications it might serve. Excellent real time and slow motion video clips of the pulsating heat pipe during operation are posted by Mertz (2000). Research on such devices has been conducted by others, but many groups are only now beginning to publish their works.

The pulsating heat pipe competes with traditional high performance heat pipes such as CPLs and LHPs more than it does mechanically pumped loops. Though it does not rely on capillary forces to circulate the working fluid, it does require capillary forces (capillary tube diameters) to form liquid slugs that act like pistons during motion and like vapor confining chambers at rest. In this respect, the pulsating heat pipe combines elements of capillary *dependent* devices and capillary *independent* oscillatory cycles. In its simplicity and elegance, the pulsating heat pipe is attractive as an economical solution to envisioned electronics cooling requirements.



A. Tamburini's T-System.

B. Akachi's Pulsating Heat Pipe.

FIGURE 2. Oscillating Thermal Systems.

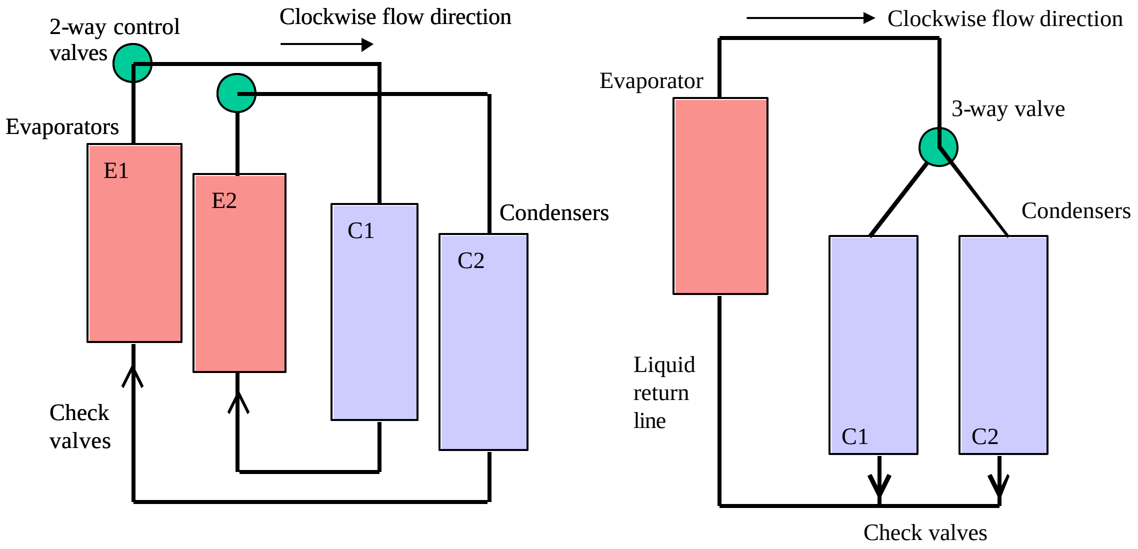
Pulse Thermal Energy Transport System (Weislogel, 1992)

The pulse thermal system proposed by Weislogel (1992) and first demonstrated by Lund et al. (1993) is similar to the Tamburini's T-System and Akachi's oscillating heat pipe (1990), only control valves (passive or other) at each of two or more coupled evaporators are used to gain precise control over the operating parameters of the loop. A schematic of this cycle is shown in FIGURE 3 where a configuration with two evaporators and two condensers is shown. Two 2-way passive or otherwise controlled valves are depicted. The valve operation is such that when one of the evaporators is in communication with its condenser, the other is sealed. A check valve at each evaporator entrance assures flow only in a clockwise direction as drawn. Operation of the loop is as follows: Heat added uniformly to the evaporators results in vaporization of liquid there. The evaporator open to its condenser decreases in pressure due to increased condensation in the condenser. The sealed evaporator simply increases in pressure. At a prescribed pressure difference (or other criterion) the valves are reversed and the high pressure vapor from the high pressure evaporator expands into its condenser where it condenses while forcing condensed liquid through the check valve into the lower pressure evaporator. The process is similar to that of Tamburini (1978) but significant control of loop performance is afforded by prescribing valve operations. For example, flow driving ΔPs can be enormous and values up to 1.8MPa (260psid) have been achieved using common refrigerants. With such large driving pressures, total heat transfer rates can be quite high as a result of high fluid circulation rates. In addition, the impact of gravity is relatively weak and effective pumping transport against >18m has been demonstrated, which is also telling of the transport distances achievable by such an approach.

Such a system may be sized to operate at a desired frequency, for a wide range of heat loads. As a result of the high driving pressure, the pulse system may be more of a competitor with mechanically pumped loops than with traditional heat pipes. The penalty for increased control and performance is additional components with moving parts, i.e. control valves. With proper fluid positioning start-up is uneventful, but conditions may be imposed where start-up requires various degrees of preconditioning. The pulse approach using control valves combines elements of capillary *independent* oscillatory cycles with mechanically controlled cycles.

Thermal Transport Oscillator (Cargille, 1993)

In a similar system to the pulse loop described above, Cargille (1993) proposed a passive 3-way valve with two condensers and one evaporator. A sketch of this cycle is provided in FIGURE 3. In this case heat added to the evaporator results in nonequilibrium vaporization of liquid there, leading to a net increase in pressure despite condensation in the connected condenser. During this time the condenser that is sealed decreases significantly in pressure as vapor from a previous pulse condenses. At a prescribed pressure difference (or other criterion) the 3-way valve is reversed and the low pressure condenser sucks vapor from the evaporator into itself, decreasing the pressure in the evaporator which in turn draws condensed liquid through the check valve from the high pressure condenser into the evaporator. The process is repeated. This cycle has the same performance potential as the pulse thermal system described above, but also possesses the same setbacks of increased system complexity, moving parts, and start-up issues.



A. Weislogel's Pulse Thermal System.

B. Cargille's Thermal Transport Oscillator.

FIGURE 3. More Oscillating Thermal Systems.

SOME PULSE LOOP CHARACTERISTICS

The oscillatory thermal systems that employ valves other than check or pressure relief valves (Garriss and Garriss (1978), Weislogel 1992, Cargille 1993) offer greater control over the cycle parameters (frequency, operating temperture), significantly larger flow driving pressures, and a wider range of operation than oscillatory cycles that do not (Tamburnini 1978, Akachi 1990, Akachi 1993). For similarly sized systems, a pulsing thermal system can transfer approximately 10-fold the heat of a high performance heat pipe (CPL), against gravity and over long distances. However, with the exception of Akachi's pulsating heat pipe, the increased complexity of the system impacts both cost and operational life, making it difficult for such pulsing systems to compete against traditional heat pipes at low powers, fluxes, and distances. The increased performance and complexity may instead argue that pulsing cycles are more competitive to mechanically pumped loops where their relative simplicity, low number of moving parts, and reduced cost might be viewed as an improvement. This is the region A on FIGURE 1.

Prototypical Pulse Loop Performance

Some quantitative performance characteristics of a prototype pulse loop will be provided here that illustrate the nature and range of operation of a given system. This particular system is selected with an evaporator section the size of a typical LHP; namely, approximately 2.54cm in diameter, 30cm in length. The transport tubing used is approximately 0.5cm throughout and the working fluid is R134a. The loop is fashioned after a cycle recently filed for patent by Weislogel (2000). All heat dissipation rates are given in Watts, all driving pressure differences are provided MPa (psid). Raw driving pressures are provided only in psid.

FIGURE 4 displays a typical passive start-up of the loop for a step heat rate of 600W. The system ΔP is set at 0.412MPa (60psid). No significant temperature overshoot at the heat source is observed. FIGURE 5 shows 60 hours of steady state data with Q = 600W and ΔP = 0.412MPa (60psid). Vapor temperature oscillation within the evaporators is seen to be approximately ±2°C, but the thermal mass of the heater yields a heater block temperature that is constant within experimental uncertainty. The slight variations in system temperature with large changes in system pressure speak to the nonequilibrium nature of the cycle. In general, in the evaporator, the liquid is saturated and the vapor is superheated. Oscillations in vapor temperature at the condenser are nearly undetectable.

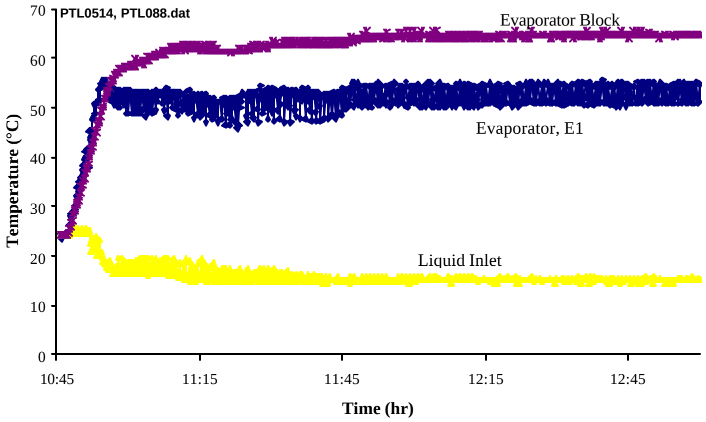


FIGURE 4. Pulse Loop Start-up: Q = 600W, ΔP = 0.412MPa (60psid).

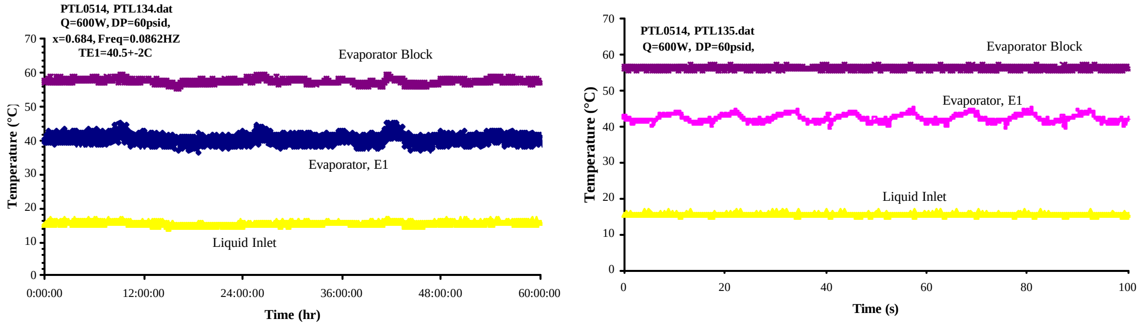


FIGURE 5. Left, 60 hr steady temperature data. Right, 100s of test at left, fluctuations are ±2°C.

FIGURE 6 and FIGURE 7 show a range of anticipated evaporator pressure difference profiles. Circulating R134a, maximum (heater configuration limited) heat rates of 2330W were achieved. Driving ΔPs of as low as 0.068MPa (10psid) and as high as 1.80MPa (262psid) were demonstrated for transport lengths (distance between evaporator and condenser) 1m to over 16m. Heater source temperature was typically less than 100°C while sink temperature was maintained at approximately 15°C for such tests. The mean operational temperatures and pressures are highly dependent on working fluid, system size and geometry. Thus, the systems may be readily tailored to the specific application. FIGURE 8 shows a sample collection of experimentally determined pulse frequency verses the calculated frequency data, the latter containing the numerous parameters (great than 24) of the system. The linear collapse of the data for a variety of test conditions shown in FIGURE 8 is indicative of the predictability of the cycle. Thermal resistance values for the cycle in the range 0.03 to 0.1°C/W have been demonstrated.

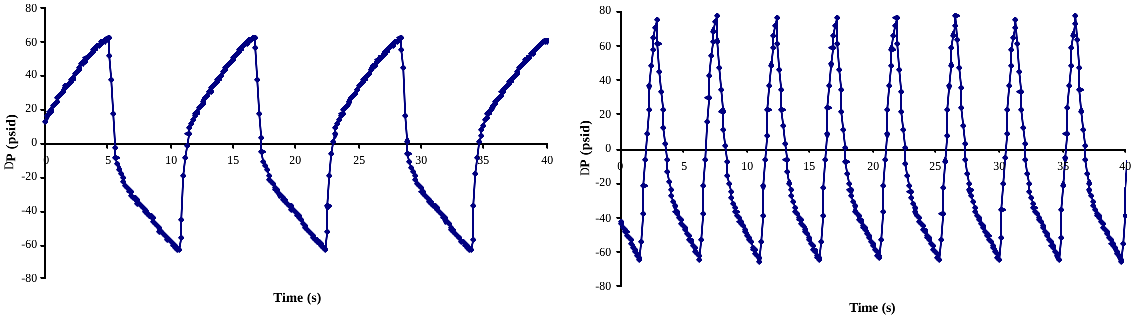


FIGURE 6. Pressure difference profile as a function of heat, ΔP = 0.412MPa (60psid), x = 0.684. Left 600W, Freq.=0.0862Hz. Right 900W, Freq. 0.206Hz.

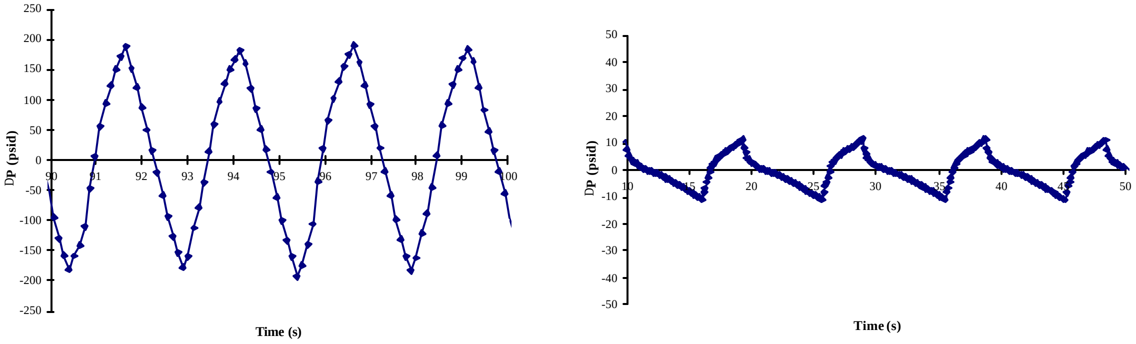


FIGURE 7. Steady evaporator pressure differentials. Left, Q = 1500W, ΔP = 1.24MPa (180psid). Right, Q = 400W, ΔP = 0.068MPa (10psid).

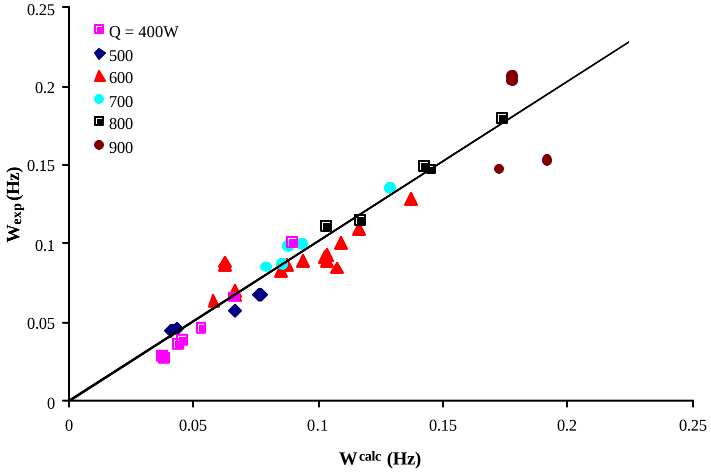


FIGURE 8. Correlated steady pulse frequencies.

OBVIOUS ATTRACTIONS AND CHALLENGES

As briefly discussed and shown above, the pulsing systems can generate large flow driving pressure gradients. These may be exploited to increase the work done on the working fluid in the form of increased circulation rate, which increases overall heat transport capability when compared to capillary pumped systems. The increased ability to do work on the fluid increases total power transport capability, but also design flexibility in that transport distances, flow resistances, and hydrostatic heads may be significantly increased. The increased ΔP comes at the price of increase system complexity—moving mechanical parts (i.e. valve, check valves). Though the valves may be extremely simple, such oscillatory thermal systems cannot achieve the simplicity and perceived reliability of tradition capillary driven heat pipes. However, exploiting the high pressure delivery, the pulsing systems can achieve transient forced convection heat transfer rates of the order achieved by some mechanically pumped loops. They can also transport the heat large distances through narrow passageways like a mechanically pumped system. But the pulse systems are semi- to completely passive, they do not require any pump power, and they have a reduced compliment of mechanical components. Perhaps most importantly for large volume production, as in meeting electronics cooling requirements, the pulsing systems do not incur the cost of the mechanically pumped system. For spacecraft applications they do not incur the weight penalty of a pump. At high enough heat loads and heat fluxes, the mechanically pumped system will be unmatched. The question posed is 'where is this threshold?' Further development of this technology is needed to answer this question.

Concerns that are certain to require closer attention are arguments concerning start-up, vibrations associated with the pulse, temperature and pressure oscillations, and loop temperature control. As regards electronics cooling applications, however, is appears that as heat load, heat flux, and transport distance requirements increase, the high performance of mechanically pumped loops will become increasingly attractive. With perfomance perhaps on par with mechanically pumped loops for this application, the pulsing systems may find a niche due to their affordability.

CONCLUSIONS

Oscillatory thermal transport cycles are reviewed that do not rely on capillary forces. Sample data is presented displaying the general characteristics of such cycles—primarily the large driving pressures achievable. These driving pressure are exploited by the loops to transfer large quantities of heat, at high fluxes, over long distances and against significant resistances and/or hydrostatic forces. The loops may outperform heat pipes (including CPLs and LHPs), but are more complex. At moderate heat loads and fluxes the loops may perform similarly to mechanically pumped loops, but are less complex as well as less expensive. They also do not require pump power.

Although the niche of the pulsing systems has yet to be identified, a competitive regime is suggested herein. The fair number of concerns regarding the oscillating transport systems will be addressed as the technologies are further developed. Applications to electronics cooling are attractive primarily because mechanically pumped systems are now in production and will become more prevalent as cooling requirements become more difficult to meet with conventional techniques. In this light the pulsing systems might be found to compete with mechanically pumped systems primarily due to equivalent performance at lower cost.

ACKNOWLEDGMENT

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REFERENCES

Akachi, H., Structure of a Heat Pipe, Foreign Pat. No. 62-155747, 1987. US Pat. No. 4,921,041, May 1, 1990.
Akachi, H., Structure of a Micro-Heat Pipe, US Pat. No. 5,219,020, June 15, 1993.
Balmer, R., *Thermodynamics*, West Publishing Company, 1990, p. 463.
Benner, S.M., Martin, M.S., Heat-Driven Pulse Pump, US Pat. No. 6,123,512, Dec. 26, 2000.
Cargille, D.B., Thermal Transport Oscillator, US Pat. No. 5,257,660, Nov. 2, 1993.
Forster, F.K., Bardell, R.L., Blanchard, A.P., Afromowitz, M.A., Sharma, N.R., Micropumps with Fixed Valves, US Pat. No. 5,876,187, March 2, 1999.

Garriss, J.E., Garriss, E., Passive Heat-Transport System, US Pat. No. 4,128,123, Dec. 5, 1978.

Jaster, Heinz, Kosky, P.G., Staub, F.W., Double Valve Vapor Push Pump, US Pat. No. 3,834,835, Sept. 10, 1974.

Lund, K., Baker, K., Weislogel, M.M., The Vapor-Pressure Loop Concept for Space Systems Heat Transport, *1st Int. Conf. on Aerospace HX Technol*., Palo Alto, CA, Feb. 1993.

Mertz, R., Pulsating Heat Pipe (PHP) Research, University of Stuttgart, http://www-ew.ike.uni-stuttgart.de/ewhsv_anw.htm.

Mudawar, I., Assessment of High-Flux Thermal Management Schemes, ITHERM 2000: International Conference on Thermal, Mechanical and Thermomechanical Phenomena in Electronic Systems, Las Vegas, NV, May 23-26, 2000.

Ottenstein, L., Thermal Management Development and Needs at Goddard Space Flight Center, *Workshop on Research Needs in Space Thermal Systems and Processes for Human Exploration of Space*, sponsor: NASA OLMSA, host: NASA Glenn Research Center, Cleveland, OH, July 725-26, 2000.

Tamburini, P., T-System: Proposal of a New Concept Heat Transport System, Proc. *3rd Int. Heat Pipe Conference*, Palo Alto, CA, 346-353, May, 1978.

Weislogel, M.M., Semi-Passive Pulse Energy Storage System, US Pat. No. 5127471, July, 7, 1992.

Weislogel, M.M., Pulse Thermal Loop, US Pat. Filed, Nov. 16, 2000.