APPLICATIONS OF HEAT PIPES IN ENERGY CONSERVATION AND RENEWABLE ENERGY BASED SYSTEMS

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ABSTRACT

In this paper, design and characteristics of different energy conservation and renewable energy based system using heat pipes as thermal control element have been discussed. Heat pipes provide two-phase reliable heat transfer system with passive operation and high effectiveness for these applications. Energy conservation system for data center cooling, agricultural products cold storage, bakery waste heat recovery and automotive dashboard cooling was achieved by using gravity assisted wickless heat pipes (or thermosyphons) and capillary pumped loop. Renewable energy based electricity generation system developed in this study utilizes thermosyphons to extract stored heat (solar pond, geothermal), to dissipate waste heat to ambient and to store waste heat into phase change materials. Heat pipe provide economical and zero greenhouse gas emission solution for these applications.

Keywords: *heat pipe, thermosyphon, capillary pumped loop, energy conservation, renewable energy, solar energy, geothermal*

1. INTRODUCTION

Thermal control is a generic need of any heat dissipation system. Heat pipes and vapor chambers have emerged as the most appropriate technology and cost effective thermal solution due to their excellent heat transfer capabilities, high efficiency and structural simplicity. Basically, the heat pipe and vapor chamber are two-phase heat transfer devices (Dunn and Reay, 1994). They involve an evacuated and sealed container with a small quantity of working fluid. One end of the container is provided with waste heat from the source, causing the contained liquid to vaporize. The vapor flows to the cold end of the container where it condenses. Since the latent heat of evaporation is much larger than the sensible heat capacity of a fluid considerable quantities of heat can be transported using these devices with a very small end to end temperature difference. For the condenser above evaporator configuration (bottom heat mode), the return of the condensate can be aided by gravity e.g. gravity assisted heat pipes or thermosyphons. While for the evaporator above condenser configuration (top heat mode) or horizontal (evaporator and condenser at same level) configuration, porous structure is lined on the inner circumference of the heat pipe to promote capillary pumping of the working fluid.

In this paper, research and development in the areas of energy conservation and renewable energy using heat pipes has been discussed.

2. ENERGY CONSERVATION SYSTEMS

In this section, different energy saving and energy recovery systems that utilizes heat pipes as one of the integral functional element are discussed.

2.1 Data Center Cooling Systems

Electric power consumed by datacenter electronics and the cooling cost associated with them are massive. Cooling cost includes the capital and operation costs associated with the active and backup cooling equipment. It has been established from past trends and future forecasts that power consumed by datacenters is one of the major loads on the electric power station and it nearly doubles every five years. Looking at more tangible comparison, power usage by US datacenters in 2011 is equivalent to power consumed by more than 11 million US houses or yearly electricity needs of whole Los Angeles city. The gravity of the problem is well understood from these massive power and cost figures. Out of the total power consumed by datacenter facility, cooling infrastructure takes up more than 50% electric load which is a huge percentage and therefore represents major cost overhead. Datacenters house mission critical computer systems and associated components for companies and organisations. Due to importance of the stored data and round clock demand on the availability of the datacenters, it is imperative to provide them with the backup power system to support computing equipment as well as their cooling systems. At present, the backup technologies used to power the cooling system are diesel generator, micro gas-turbine or electric battery based. It is evident that most of the present backup systems use non-renewable energy sources like oil or gas to produce power which degrades earth's atmosphere by greenhouse gas emission. Batteries need electricity from grid to maintain their charge level and produce environmental contamination during their disposal. Due to the massive size of datacenters, it can be argued that backup cooling systems size and extent of sources (oil, gas or electricity) needed to power them is also substantial. It should be noted that it is not only the running cost of the non-renewable backup systems but also the environmental

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damage caused by them is considerable. Based on the above discussion, it can be concluded that propositions of energy conservation system for datacenters can save substantial electricity (thus running costs) and greenhouse gas emissions to the environment.

In the present section, two types of energy conservation systems based on heat pipes (Singh et al., 2010) are proposed for datacenters: 1) Heat pipe heat exchanger (HPHE) pre-cooler for datacenter chiller and 2) Heat pipe based ice storage system for datacenter emergency cooling. Both the systems utilize thermal diode characteristics of passively operating heat pipes, as shown in Figure 1, to capture cold ambient energy for cooling purposes. The thermosyphon can extract heat from the high temperature storage media to low temperature ambient by means of continuous evaporation-condensation process. In other words, the thermosyphon can only transfer heat in one direction i.e. when operating in the bottom heat mode (evaporator below condenser).

Fig. 1 Thermal diode character of heat pipe

2.1.1 Heat Pipe Heat Exchanger Pre-Cooler

In the proposed pre-cooler system the heat pipes transfers the cold energy from the ambient to the coolant flowing inside the pre-cooler duct in real-time thereby cooling it by certain degrees, depending on the pre-cooler design and ambient air temperature, before it enters chiller to achieve the designed cold plate inlet temperature for datacenter servers.

Fig. 2 Datacenter facility with HPHE pre-cooler

Figure 2 shows the schematic of the data center facility with the pre-cooler integrated between the data center racks and chiller (Wu et al., 2011). The coolant used to extract heat from the data center server is mixture of water and propylene glycol (50-50% by volume). For the 120 kW data center, the pre-cooler has been designed to handle one-fourth (30 kW) of the heat load for 30 °C ambient temperature which represents the upper limit of summer time temperature for Poughkeepsie, New York.

Fig. 3 Dependence of HPHE pre-cooler size and heating effectiveness on designed ambient air temperature

Fig. 4 Yearly heat load handling capability of the HPHE Pre-cooler

HPHE Pre-cooler has 118 heat pipes with 38% heating effectiveness for coolant inlet and outlet temperatures of 51 °C and 43 °C respectively. The evaporator and condenser length of the heat pipes is 1 and 2 m respectively with 90 mm (L) x 90 mm (W) x 0.5 mm (T) fins on condenser side. The air velocity of 1.8 m/s provides forced air cooling to the condenser portion. As the fan velocity is very close to yearly average wind speed (1.7 m/s) therefore power consumed by the air movers will be very low.

The size and performance of the HPHE pre-cooler is very dependent on the designed ambient air temperature (in this case 30 °C). Figure 3 presents the effect of the ambient temperature on the HPHE size and effectiveness. It should be noted that the designed temperature is location dependent therefore to install the similar capacity heat exchanger at hotter location (e.g. Singapore), the size will be much larger. In Figure 4, the yearly load handling capability of the HPHE pre-cooler has been estimated. It is evident that the exchanger is able to dissipate the designed heat load (30 kW) through the year with capacity higher than designed value during most time of the year, except peak summer season. The main cost associated with the HPHE Pre-cooler is heat pipe and tank cost. Based on the yearly performance of the pre-cooler, it is estimated that the payback period for the designed HPHE pre-cooler will be around 2.8 years.

2.1.2 Ice Storage Emergency Cooling System

For ice storage system, the cold energy from the below zero winter ambient is captured by heat pipe and used to convert storage water into ice. This ice is stored throughout the year inside the well-insulated underground storage as emergency cooling support for datacenter facility in case of any electricity disruption to the main chiller equipment.

In the normal operational mode, data center is cooled by using chiller and cooling tower arrangement as shown in Figure 5. During the unavailability of the chiller due to system failure or power disruption, the three way control valves can channel the hot coolant coming from the data center to the underground ice storage. The ice storage system comprise of number of ice storage module which is made up of thermosyphon as ice charging component using sub-zero ambient temperature and coil heat exchanger as ice discharging component through which hot water propylene (PP) glycol mixture (used as coolant in data center facility) can be pumped whenever the failure of the primary cooling equipment occurs. As an alternative arrangement, the individual coil heat exchanger for each heat pipe can be replaced with the tube-type single heat exchanger, with inlet and outlet headers, that can extract cold energy from multiple ice generation spots (around heat pipe circumference). During ice storage charging process, the individual thermosyphon will extract heat from the high temperature storage media (water in underground storage tank) to sub-zero winter ambient resulting in phase change of water to ice. Such a system will be feasible for places where ambient temperature goes below zero degrees Celsius during winter season. In order to avoid melting of ice during hot climatic conditions, the ice storage tank should be installed underground and provided with thick layer of insulation all around. Depending on the size of the data center and number of hours of failure support required, number of ice storage module can be combined together to form an overall backup cooling system as shown in Figure 5. In the present case, emergency ice storage system is designed to support data center with 1 MW output heat load capacity for 6 hours (Wu et al., 2010; Wu et al., 2010; Singh et al., 2011).

Fig. 5 Data center facility with Heat Pipe Ice Storage System

Detailed theoretical analysis was conducted to estimate the yearly ice generation capacity of individual thermosyphon module for the given heat pipe dimensions and climatic conditions of the chosen location. In the current analysis, heat pipe consists of stainless steel tube with 50.8 mm outer diameter, 3 m evaporator length and 3 m condenser length. The condenser portion consists of 300 aluminium fins with 300 mm x 300 mm area and 1 mm thickness. R134a was used as working fluid due to its superior thermal performance at lower temperatures as compare to other heat pipe working fluids. The climatic conditions for Poughkeepsie in New York were used to estimate thermosyphon ice generation capability.

Figure 6 presents the results of the analysis which shows that approximately 952 kg of ice per winter season can be produced by individual heat pipe module. Total of 68 heat pipes will be needed for 1MW data center emergency support for 6 hours assuming not more than one failure per year. Depending on the failure history for particular location, the heat pipe system can be sized with certain factor of safety. Mechanical work constitutes the main cost related to the ice storage system. Based on the cost analysis, it is estimated that

the ice storage system will cost from 0.3 to 0.68 \$/W as compared to the existing data center backup systems (diesel/micro gas turbine generator or fuel cell) which cost around 0.5 to 1 \$/W.

Fig. 6 Ice generation capability of single heat pipe module

The novel concept of heat-pipe based ice-storage system was experimentally tested at Fujikura test facility at Aomori in Japan. Figure 7 (a) presents the details of the experimental test facility showing the condenser portion of heat pipe exposed to ambient with evaporator portion installed underground. The heat pipe module was made of stainless steel and R134a was used the working fluid. In order to minimize heat leaks from the ambient, the evaporator section of the thermosyphon along with the storage tank was installed underground level and the outer surface of the tank was insulated with thick layer of fibre glass insulation. The experiment was carried out for a winter period of 25 days during which the ambient temperature was below zero degrees Celsius. Figure 7 (b) shows the results of the visual inspection that was done at the end of the test period which clearly validate the proposed concept. Based on the measured temperatures, it was estimated that approximately 113 kg of ice was produced during the test period.

Fig. 7 (a) Experimental set-up for proof of concept (b) Ice generation around the thermosyphon and inside ice storage tank

2.2 Cold Energy Storage for Agriculture Products

The cold storage system, as described above, has also been investigated by Fujikura for agriculture product preservation for longer time period (Mashiko et al., 1989, Tsuchiya et al., 1991). Figure 8 shows the concept of using a heat pipe to collect cold energy in the winter season and storing underground to create a permafrost system for storage of agricultural products throughout the year. The pilot plant shown was built in Hokkaido, Japan, where the freezing index is above 400°C per day. Figure 9 plots the freezing index versus warmest month average temperature for different locations throughout the world. Freezing index (FI) is defined as the number of below zero degree Celsius days per year. For example, if a place has an average temperature of -4 °C for 100 days in a year then the freezing index of the place is 400 °C.days. It can be used to estimate the ice forming potential of the location.

The plant consists of 216 stainless steel heat pipes of 46 mm diameter and 12 m length installed around the store room at 0.5 m intervals. Size of the store room was 6 m long x 3.6 m wide x 3.6 m high. The heat pipe evaporator is buried in the soil, and the condenser is exposed and cooled by natural air. In winter season, the cold energy system was able to create 2-m-thick frozen soil layer. The permafrost soil layer temperature was kept at -1°C even in hot month of July which helped to keep store room temperature about 3 to 4°C and humidity over 85% throughout the summer period. The overall cold energy system is fully passive, i.e., there are no moving parts, no electrical consumption, and hence it is a reliable and maintenance free system.

Fig. 8 Permafrost cold storage system for agriculture products

Fig. 9 Freezing index versus warmest month average temperature for different locations

The proposed system provides sustainable and environmental friendly approach by saving electricity for stored food cold storages. Such a system is economically viable in cold regions with freezing index greater than or equal to $400^{\circ}\text{C}\cdot\text{days}$.

2.3 Bakery Waste Heat Recovery

A waste recovery system based on the heat pipe heat exchanger (HPHE) was designed and implemented by RMIT University at Buttercup Bakery (Dube et al., 2004). In this system, a HPHE was

used to recover waste heat from the high temperature baking oven for heating the low temperature proofing oven. In Figure 10, details of the waste heat recovery system are presented. The temperature of the baking oven's flue gases ranged from 300 – 350°C and available waste heat was 70 – 80 kW. For the proofing oven the energy requirement was between 20 – 45 kW. With an air velocity of 1.5 m/s and heat exchanger effectiveness of approximately 65%, the HPHE based waste heat recovery system was able to supply all the heat needed by the proofing oven, thus eliminating the need for natural gas heating.

Fig. 10 Bench-type prototype of heat pipe heat exchanger based waste heat recovery system for bakery (Dube et al., 2004)

The Buttercup heat exchanger was manufactured locally from steel pipes charged with distilled water. Figure 11 presents a schematic of the HPHE. When hot exhaust gases from the baking oven come in contact with the HPHE evaporator containing liquid, the liquid boils and vapour is transferred to the condenser portion by natural convection where it condenses, thereby transferring energy as latent heat. The heat extracted is fed directly into the proofing oven, conserving energy while providing uncontaminated air. The predicted annual waste heat recovery from one eight hour shift working six days a week was estimated to be approximately 500 GJ.

Fig. 11 Schematic of Heat Pipe Heat Exchanger (Dube et al., 2004)

The total cost of manufacturing and constructing the 50 kW HPHE prototype, excluding data monitoring operations, was approximately \$10,000. Based on annual saving of 500 GJ per year for one shift, cost of natural gas at \$4 per GJ, and boiler efficiency of 70%, the payback period is less than 3.5 years. For a three shift operation, the payback period is reduced to 1.5 years. Additional benefits includes more efficient operation of the boilers in terms of lower steam requirement, less gas burnt, and less maintenance on the boilers.

The research work investigated in detail the effect of the non-condensable gases (NCGs) on the heating effectiveness of the loop thermosiphon heat exchanger (LTHE). It was observed from experiments that the effectiveness based on the condenser heat output fell from 26.3% to 20.5%, while the effectiveness based on evaporator heat input decreased from 33.2% to 26.9%. Four reservoirs were installed at different locations of the condenser, as shown in Figure 10 to determine the storage location of the NCG. In the experiment, the quantity of the NCG was increased inside the LTHE system by inducing air through opening the drain valve. As an outcome, the temperature difference between vapour and the reservoir surface for each reservoir increased accordingly due to increases in the thermal resistance offered by NCG. It was observed that the temperature difference between condenser vapour and reservoir surface and thus amount of induced NCG was highest in reservoir A followed by reservoir B, which in turn was always higher than those in reservoirs C and D. It was therefore propose on the basis of this experimental study that the most suitable location for a gas reservoir to collect NCG in an LTHE is on the bottom header of the condenser where the condensate exits to the evaporator by gravity effect through adiabatic pipe.

2.4 Automotive Dashboard Cooling

The capillary pumped loop (CPL) (Singh et al., 2004) prototype was designed and tested to study its unique operational and temperature control characteristics. Such a system can be used in terrestrial applications including automobile engine cooling and space applications including electronic cooling in satellites (Singh et al., 2004). CPL has the potential to transfer large heat loads over long distances with small temperature differential and no external power requirements except for the thermal control of the reservoir (Singh et al., 2004). In this investigation, the intended application was cooling of a car dashboard which was being heated by radiation heat transfer from the engine. The proposed system will help in the passive removal of the heat from the car dashboard which will reduce the demand on the car air conditioner thereby reducing energy expended on cabin cooling (Singh et al., 2004).

Fig. 12 CPL with aluminium evaporator and acetone as working fluid

Figure 12 shows various components of the CPL experimental model. The main design characteristics of the CPL model are given in Table 1. Aluminium was chosen as the evaporator material for its light weight which was a requirement of the design. In the present tests, the CPL test bed was operated in the horizontal configurations. High grade Acetone, a saturated aliphatic ketone, with a minimum assay level of 99.95% was used as working fluid due to its chemical compatibility with aluminium, stainless steel and UHMW polyethylene material at low as well as with high temperatures. Acetone is also very cheap to obtain in a high purity state and the working pressures are not very high.

Table 1 Main design parameters of CPL prototype

Parameter	Value
Evaporator shape	Cylindrical shape
Evaporator material	Aluminum
Evaporator ID/OD (mm)	28.5/31.5
Evaporator length (mm)	500
Vapour removal channels (Groove top width / Groove bottom width / Groove depth) (mm)	1.5/2/2
Wick material	UHMW polyethylene
Wick ID/OD (mm)	12/25
Wick length (mm)	440
Wick mean pore radius (μm)	20
Wick porosity	50%
CPL material (except evaporator)	Stainless steel
Reservoir volume (litre)	0.75
Flow lines ID/OD (mm)	4.5/6.4
Vapour/liquid line length (mm)	1000
Condenser line length (mm)	4300
Condenser type	Forced air, coil & plate type
Condenser material	Cu plate, Stainless steel tubing
Working fluid	Acetone

The CPL presented very good response to changing heat loads and reached steady state within a short time period. It was observed that the CPL was able to maintain the temperature at the evaporator within 52 to 57 °C for input heat load of 30 to 100 W and reservoir set temperature of 40 °C. For input heat load of 100 to 210 W and reservoir set temperature of 50 °C, the evaporator temperature was maintained within 55 to 58 °C. It is evident from the outcomes that the CPL exhibited very efficient control over the operating temperature of the evaporator for the entire range of input power. These tests were conducted on the CPL with the condenser maintained at room temperature by air cooling using forced convection. Figure 13 presents the total thermal resistance from CPL evaporator external wall to ambient air over range of applied heat load. For total thermal resistance, a minimum value of 0.19 °C/W was achieved at the maximum heat load of 210 W and at evaporator temperature of 58 °C.

Fig. 13 Variation of the total thermal resistance with the applied heat load

The current investigation helps in the classification of capillary pumped loops as efficient thermal management systems that can be used for the precise control of the source temperature within limits which are decided by the system design and heat transfer requirements.

3. RENEWABLE ENERGY SYSTEMS

In this section, different solar and geothermal based renewable energy systems that utilise heat pipes as an integral element are discussed.

3.1 Heat Pipe Turbine

A new concept for power generation from solar, geothermal or other available low grade heat sources using a Heat Pipe Turbine or a Thermosyphon Rankine Engine (TSR) was developed and tested (Nguyen et al., 1999). The basis of the engine is the modified thermosyphon cycle, with its excellent heat and mass transfer characteristics, which incorporates a turbine in the adiabatic region. Figure 14 shows the details of the heat pipe turbine with that consists of closed vertical cylinder with an evaporator, an insulated (adiabatic) section and a condenser. The turbine is placed in the upper end between the evaporator and condenser sections, and a plate is installed to separate the high pressure region from the low pressure region in the condenser. Conversion of the enthalpy difference to kinetic energy is achieved through the nozzle. The mechanical energy developed by the turbine can be converted to electrical energy by direct coupling to an electrical generator.

Fig. 14 Heat Pipe Turbine (Nguyen et al., 1999)

Development of the heat pipe turbine was conducted by fabricating and testing a series of prototypes in the RMIT Laboratories. The aim of the work was to optimise the performance of each successive prototype in order to simplify manufacture and to increase the power output. The first prototype turbine rotated at 600 to 800 rpm but only very low power output was measured. The second prototype was equipped with an impulse turbine and the heat input was specified as 2 kW. The rotational speed of the turbine was up to 3400 rpm, but the power output was still low. In the third prototype the reaction turbine was introduced. The turbine was produced as a hollow disc, including two convergent-divergent nozzles at its periphery. The diameter of the heat pipe was 16 cm and its height 3.15 m. An electrical power output of 5.5 W at 4788 rpm was obtained from a heat input of 4.4 kW. The fourth prototype consisted of a cylinder of height 2.8 m and a diameter of 0.5 m. The turbine configuration was the same as in the third prototype. The heat input was 10 kW and electrical power output of 100 W was obtained at 6000 rpm. Although the efficiency of the proposed heat pipe turbine is low, it has the capability of utilising very low grade heat and converts it into useful electricity.

3.2 Geothermal Heat Extraction

A large scale loop type heat pipe for extraction of geothermal energy was developed at Fujikura (Mashiko et al., 1994). The heat pipe of 150 mm outer diameter and 150 m length was manufactured and installed in a geothermal well, with 100 to 150 °C temperature, located at Kyushu Island in Japan, as shown by schematic in Figure 15. Conventional heat pipes would not work properly in the present application by simply enlarging the heat pipe diameter and length because the heat load will cause entrainment and flooding phenomena within the heat pipe. With a long heat pipe heat-transfer performance would deteriorate due to the difficulty of maintaining uniform liquid film throughout the length of the heat pipe. To address these heat transfer issues in a longer vertical evaporator section, an innovative design using liquid feeding tubes with showering nozzles was used in the evaporator.

Cross section of evaporator section showing the liquid holding wick and liquid return lines with spurting nozzle are shown in Figure 16. The liquid is fed through a feed tube having a number of nozzles to spray the liquid onto the inner surface of the evaporator of the heat pipe. The spraying of liquid on the evaporator enhances the evaporation heat transfer of the heat pipe. This new concept helps to provide twofold advantage firstly by separating the liquid and vapour phases to avoid entrainment losses and secondly by providing uniform liquid film in the evaporator section for high heat transfer rate.

Fig. 15 Large scale heat pipe for extracting geothermal heat

In the trial tests, the heat pipe was able to continuously extract 90 kW heat at the working temperature of 80 °C. Heat flux at the evaporator was of the order of 3000 W/m² (Mochizuki et al., 1995). Figure 15 shows the schematic of the demonstration rig. Due to the larger length of the evaporator, the novel concept of the showering nozzle was used to spread thin layer of the liquid over the inside of the evaporator tube. Liquid was returned using gravity effect by multiple return tubes. The nozzles spray the working fluid under hydrostatic pressure due to height of liquid column on the wick structure which provided high evaporative heat transfer with lower thermal resistance.

Fig. 16 Heat pipe evaporator cross section (Mashiko et al., 1994)

Figure 17 presents the experimental results on the heat extraction obtained by large scale heat pipe unit. It is noted from the graph that approx. 65 kW of continuous heat was extracted at a flow rate of 1.2 L/min and heat pipe working temperature of approx. 70 °C. It is estimated that over 100 kW can be obtained at a working temperature of 100 °C by using this large scale heat pipe prototype.

Fig. 17 Test results for geothermal heat extraction by a large-scale heat pipe

3.3 Thermosyphon Based Thermo Electric Generation System for Solar Ponds

3.3.1 Introduction

Salinity gradient solar ponds are large bodies of water that act as solar collectors and heat storages. Solar pond is a simple and low cost solar energy system which collects solar radiation and stores it as thermal energy for a relative longer period of time. When solar radiation penetrates through the solar pond surface, the infrared radiation component is first absorbed in the surface mixed layer or upper convective zone. However, this heat is lost to the atmosphere through convection and radiation. The remaining radiation will subsequently be absorbed in the non-convective zone and lower convective before the last of the radiation reaches the bottom of the pond. In these ponds the solution is heavier in the lower region because of higher salt concentration. As a result the natural convection that takes place in normal ponds is suppressed. Solar radiation penetrating to the bottom region is thus absorbed there, and temperature of this region rises substantially since there is no heat loss due to convection. The temperature difference created between the top and the bottom of the solar ponds can be as high as 50°C to 60°C. The collected and stored heat can be extracted and used for industrial process heat, space heating, and even power generation (Akbarzadeh et al., 2005).

A typical solar pond consists of three regions, the Upper Convective Zone (UCZ), the Non Convective Zone (NCZ) and the Lower Convective Zone (LCZ) as shown in Figure 18. The upper convective zone is the topmost layer of the solar pond. It is a relatively

thin layer that consist almost fresh water. The non-convective zone is just below the upper convective zone and has an increasing concentration gradient with respect to the pond depth and relative to the upper convective zone. It also acts as thermal insulation for the bottom layer. The LCZ has highest salt concentration without any salinity gradient. In solar ponds, if the concentration gradient of the NCZ is great enough, no convective current will occur in this region, and the energy absorbed in the bottom of the pond will be stored in the LCZ. While construction and maintenance of a solar pond of this size is not a problem, the conversion of thermal energy to power is a difficult challenge. Conventional heat engines have too many moving parts and are complex. They are very expensive (15000 to 20,000 \$/kW) in these small sizes (less than 0.5 kW for the above example) and difficult to maintain.

Fig. 18 Salinity gradient solar pond

In the present research it is shown that by combining thermosyphons and thermoelectric cells, it would be possible to utilize the temperature difference existing between the top and the bottom of a solar pond and produce electric power in a fully passive way, i.e. no moving parts. In such a scheme, the heat is transferred by the thermosyphon from the lower region of the pond to the 'hot' side of thermoelectric cells which maintains a good thermal contact with the top of the thermosyphon tube. The 'cold' sides of the cells are in contact with the cold environment of the top layer of the solar pond.

The above proposal utilizes thermosyphons, which are highly effective devices for heat transfer (Dunn and Reay, 1994), and thermo electric cells (TECs), which can effectively convert a temperature difference to electric potential and generate power (Rowe, 1994). Both of these devices do not have any macro scale moving parts and are thus fully passive. Although at present, the efficiency of conversion of heat to electricity by thermoelectric cells is still low (2% for a 50 °C temperature difference) and at its best is 10 to 20% of the Carnot efficiency for the same temperature difference, the availability of the cells and their simplicity suggest that these devices may be very suitable candidates as small and simple energy converters in applications such as small solar ponds. Of course the developments in recent years in semiconductor materials for thermoelectric cells and the resultant improvements in their efficiency are promising indications that these cells have good potential to be economically viable candidates for conversion of low-grade heat (produced in most solar collectors) into electricity. It should be noted here that the technology for the manufacture of thermosyphons for application such as above is fully developed and thermosyphons of different sizes can be easily manufactured requiring only a moderate amount of skill.

In remote areas, where the electric grid is not available and the sun shines year round, combined power generation modules based on the small scale solar pond, thermosyphon and Thermo Electric Generator (TEG) is one of the viable candidates for providing daily electricity demand in such areas. A TEG has the advantage that it can operate from a low grade heat source such as waste heat energy. It is also attractive as a means of converting solar energy into electricity. The schematic diagram of the TEG is shown in Figure 19. It consists of two dissimilar materials, n-type and p-type semiconductors, connected electrically in series and thermally in parallel.

Heat is supplied at hot side while the other end is maintained at a lower temperature by a heat sink. As a result of the temperature difference, the current flow through an external load resistance. The power output depends on the temperature difference, the properties of the semiconductor materials and the external load resistance.

Fig. 19 Schematic diagram of the Thermo Electric Cell

3.3.2 Proposed System Description

The schematic of the proposed power generation unit is shown in Figure 20 (Tundee et al., 2008). Here, the thermosyphon, which is basically an evacuated copper tube charged with water as the working fluid, is held vertically and is long enough to connect the bottom convective zone of the pond to the top convective zone.

Fig. 20 Schematic of combined thermosyphon and thermoelectric cells as electricity generation module

In Figure 20, a typical temperature profile for a solar pond is also given. It is seen from this profile that the temperature is low and uniform in the top zone. This uniformity is caused by the wind induced wave actions near the top. In this zone the temperature follows closely the daily average ambient temperature. In the middle layer of the pond, where the convective currents are suppressed because of the existence of a strong salinity gradient, the temperature rises continuously until it reaches a maximum near the interface with the lower layer. In the bottom layer the temperature is also uniform as in the top layer. The mixing in the bottom zone is primarily due to the absorption of solar radiation by the dark bottom of the pond, and the convection currents

that are thereby induced. Therefore the non convecting middle layer, which is sometimes called the gradient layer, separates the upper zone (cold) and the bottom zone (hot). Some typical thicknesses for these three layers are: 0.2m to 0.6m for the top layer, 1m to 2m for the middle layer, and 0.5m to 5 m for the bottom layer. The middle section of the thermosyphon (adiabatic section) is insulated to prevent heat losses from the thermosyphon to the surrounding water in the gradient layer, which is at a lower temperature than the bottom layer. The thermoelectric cells are attached to the top part of the thermosyphon (condenser section) with a good thermal bond. The other side of the cells is cooled by the cold and convective currents available in the top convective zone. Therefore the required temperature difference for power generation is created on the two sides of the thermoelectric cells.

The working fluid is continuously evaporated in the evaporator and the resulting vapor travels upward because of the lower pressure in the condenser section caused by a lower temperature. The vapor is then condensed releasing its latent heat, which is transferred to the sides of the thermo electric cells attached to the thermosyphon. The resulting condensate travels downward because of gravity. As a result the two sides of the cells are maintained at different temperatures, and hence an electric potential difference is generated across the cell. On applying an external electric load, an electric current is produced and electric power generated. The produced electric power can be directly used for applications requiring direct current, or converted into alternating current if needed. The electric energy thus produced can be also continuously stored in batteries to provide power to intermittent loads, which may have a higher demand than the capacity of the thermoelectric cells for a period of time.

3.3.3 Laboratory Testing and Results

Figure 21 presents the schematic of the TTM prototype fabricated for lab testing and system characterisation.

Fig. 21 Tests assembly to simulate the operation of the combined thermosyphon and thermoelectric cells under solar pond conditions

Indoor laboratory tests were carried out on a combined system of thermosyphon and thermoelectric cells to simulate the operation of the

system when installed in the solar pond. For these purpose a thermosyphon was made from a 100 mm diameter copper tube with a total length of 2m. The thermosyphon was charged with water as the working fluid. As seen in the Figure 21, the thermoelectric cells were attached to the top part of the thermosyphon. This is the part that will be in the top convective zone of the solar pond which is the heat sink. Suitable thermoelectric cells that can be used for power generation were obtained from Kryotherm Ltd, Russia [17]. These cells were made from Bismuth Telluride (Bi_2Te_3) with 40 mm by 40 mm area and 3.9 mm thickness. Sixteen TEG cells were individually tested by subjecting them to temperature differences by heating one side and cooling the opposite side. As shown in the Figure 21, the lower section is heated by a hot water reservoir. This results in the heating of the inner side of the thermoelectric cells through the thermosyphon. The outer sides of the cells are cooled by flowing water that simulates the cooling effect of the top layer of the solar pond.

Figure 22 shows the test results on the Thermosyphon Thermoelectric Module (TTM) for different heat source temperature (i.e. hot water bath temperature) (Singh et al., 2011). On the figure, the temperature difference achieved across the TEG, when the water bath temperature is changed from 50 to 90 °C, is mentioned (with the top curve corresponding to the 90 °C). It can be seen from the graph that the power generated by the TTM module increase with the increase in the temperature across TEG which tracks the increase in the temperature of the hot water bath. Here, the temperature of the hot water bath simulates the temperature of the bottom dense layer of the solar pond.

For each setting of evaporator temperature, the maximum power point for the TTM system was achieved by changing the external resistance (or load). For evaporator surface temperature of 90°C, maximum power of 3.2 W was generated by the TEG units. It should be noted that the temperature across the TEG does not increase in the same proportion as the water bath temperature due to the limited cooling on the cold side of the TEG. In this case, the open-circuit voltage and the short circuit current values of 26 V and 0.4 A respectively were obtained with the maximum power point of 3.2 W obtained at 13.4 V and 0.24 A.

Fig. 22 Dependence of output power on ΔT across TEG and external resistive load

By using a number of the above cells in series one would be able to design a TTM suitable for battery charging (12 V) using the heat from the bottom of the solar pond. The output power of the unit can be between 10W to 20W under normal conditions of the pond. A number of the above TTMs can form the power supply system for a small house. For example for a small energy-efficient house, the needed electrical energy can be as low as 2 kWh per day. Since the TTM works day and night total number of 10 units would be required to power the remotely located house without access to grid power.

3.4 Concentrated Solar Photo Voltaic Generation with Heat Pipe Cooling

3.4.1 Introduction

Concentrated photo-voltaic (CPV) technology offers incentives in terms of high conversion efficiency, semiconductor material savings and low unit electricity cost (\$/W) as compared to flat-plate photo-voltaic modules. Thermal control of photo-voltaic cells is very critical in order to reduce temperature-related performance drop and to avoid material degradation. For every 1 °C drop in temperature of the PV cells, their energy conversion efficiency, in general, increases by 0.5%. This section outlines the concepts of using heat pipe based passive cooling system for solar CPV module.

One of the critical issues faced by the CPV systems is the effective cooling of the solar cells under high concentration ratio. In the present work, passive heat pipe system has been proposed for the thermal control of the CPV system. The proposed system will have high reliability and will not use any active power for its operation thereby paving the way to sustainability.

3.4.2 System Description

A solar concentrator with concentration ratio of 20 suns was combined with photovoltaic cells and thermosyphon cooling module (Akbarzadeh and Wadowski, 1996) to fabricate a CPV system, as shown in Figure 23.

Fig. 23 Solar concentrating photo voltaic module with heat pipe cooling system

The trough type concentrator with 1 m length and 0.8 m width was aligned in east-west direction. A special reflective film by 3M was lined on concentrator surface to provide high reflectance of solar radiation from concentrator to solar cells. Polycrystalline solar cells with dimension of 25 mm by 20 mm were installed on both sides of the evaporating surface of the cooling system, as seen in Figure 23. These solar cells were electrically-connected in series to achieve useful output voltage. The thermosyphon module was made of copper with flattened evaporator section for installing solar cells and using R-11 as the working fluid. In this case, R-11 was used due to the requirement to maintain the solar cell to temperature as close to ambient as possible. The condenser section of the thermosyphon consists of aluminium fins throughout its length, for dissipating heat to ambient by natural convection.

3.4.3 Test Results

The system was tested on a sunny day and produced an open circuit voltage of 34 V, short circuit current of 1.7 A and maximum electric power output of 20.6 W. The system was exposed to solar radiation for a period of 4 hours during which the temperature at the solar cell surface did not exceed 46°C. Under one sun, the system produced 2 W which is only one-tenth of the concentrating system thereby proving

Based on this study, large scale CPV module using heat pipe cooling system is under investigation in Fujikura. Figure 24 presents design of one such module. The design system can have concentration ratio as high as 125. It should be noted that the proposed system can also replace PV cells with the thermoelectric cells.

3.5.1 Introduction

To overcome this drawback, phase change material thermal storage concept has been proposed in this investigation (Tan et al., 2010). For transporting large amount of heat from the cold side of TECs to the phase change material (PCM) thermal storage tank, two-phase closed thermosyphons were used in the design. Combination of thermosyphons and PCM will provide a reliable thermal control system for concentrated thermal electric system by maintaining maximum cell operation temperature below 250°C limit. The objective of the present study is to assess the thermal performance of the concentrated thermoelectric generator system using thermosyphons as heat transporting devices.

In this proposed design, as shown in Figure 25, three thermosyphons or gravity-assisted wickless heat pipe are implemented into the CTEG-

The solar energy is concentrated on the hot side of the thermoelectric cells by Fresnel lens concentrator. In this case, two copper blocks were used to sandwich the thermoelectric cell (TEC) to provide necessary temperature difference across TECs. The evaporator sections of two thermosyphons were inserted into cold side copper block to acquire excess heat and transfer it to the PCM storage, in which the finned condenser sections of thermosyphons are installed. As mentioned earlier, one thermosyphon has its evaporator section embedded in the PCM tank for removing stored heat from the melted PCM to the cold night time surroundings. It should be noted that thermosyphon acts as thermal diode and allow heat transfer only in one direction i.e. in bottom heat mode (evaporator below condenser configuration).

Figure 26 presents the experimental prototype of thermo electric generator with thermosyphons and PCM storage tank. In the lab prototype, only two thermosyphons (one primary and one secondary) were used to validate the passive cooling concept. For the primary thermosyphon, only condenser section is finned whereas for secondary thermosyphon both condenser and evaporator are finned. The thermoelectric cells, as shown in Figure 27, were sandwiched between two copper blocks, one provided with holes for installing heating rods and the other provided with holes to install thermosyphon evaporators. For lab testing, electric cartridge heaters were used as heat simulators. Figure 28 presents the Fresnel lens concentrator which will be used to test the proposed system.

Fig. 27 TEC sub-assembly with heating and cooling arrangement

Fig. 28 Fresnel lens concentrator with the tracking system and structural support

The PCM used in the thermal storage tank is paraffin wax. It is non-corrosive, readily off-the-shelves and has relatively high latent heat storage capacity (140kJ/kg) which make it attractive to be used in the thermal storage. Table 2 shows the thermo physical properties of paraffin wax.

Table 2. Thermo-physical properties of proposed paraffin wax.

Description	Value
Melting temperature ($^{\circ}\text{C}$)	47
Solid density (kg/m^3)	880
Liquid density (kg/m^3)	760
Latent heat capacity (kJ/kg)	140
Specific heat capacity (solid/liquid) (kJ/kg.K)	2.4/1.8
Thermal conductivity (solid) (W/m.K)	0.2

It is noted that paraffin wax has very low thermal conductivity (0.2 W/m.K) which can affect the heat transfer performance of the thermosyphons. In order to overcome this issue, thermosyphon sections installed inside PCM tank were adequately finned. These fins help to improve the effective thermal conductivity of the PCM. All gravity-assisted thermosyphons were tilted at 5° angles to facilitate the return of condensate to the evaporator. In addition to this, PCM storage tank should be elevated above the TEC modules to ensure that thermosyphon condenser section is above the evaporator section.

A numerical model was developed in MATLAB to characterize the performance of the CTEG-PCM system (Tan et al., 2011). Figure 29 shows the overall thermal performance of CTEG-PCM system under concentration of 75 suns. The maximum TEC hot side temperature in the TEG module reached 240°C . Hence, for the maximum solar concentration ratio (CR) of 75, the aperture diameter of the Fresnel lens should be sized to 560 mm. The TEC cold side temperature can be maintained at 88°C which corresponds to a constant temperature difference of 152°C across TEC modules. As evident from graph in Figure 29, the dissipated heat is absorbed by the PCM as sensible heat for first 200 seconds and thereafter latent heat absorption upon reaching the melting point (47°C). The PCM temperatures (T9 and T10) located at 10mm and 50mm above the primary thermosyphon condenser are designated to capture melting performance during heat absorption. T9 first rises gradually and then become constant during melting process. T10 remained unchanged during the test period of 30 minutes due to the low PCM thermal conductivity. Figure 30 presents the effect of concentration ratio on the temperature difference across two TECs and output power. CR of 75 gives maximum output power of 9.5 W at temperature difference of 150 across TEC. Test runs will be conducted on the fabricated system to validate the numerical model. The proposed system can utilize photonic energy directly from sun, if the TECs are replaced by PV cells.

Fig. 29 Thermal performance of CTEG-PCM system (CR = 75).

Fig. 30 Temperature difference and output power from 2 TECs under different concentration ratio

4. CONCLUSIONS

The paper can be summarized as:

- Heat pipe based passive systems can provide reliable and effective thermal control for energy conservation, energy recovery and renewable energy applications.
- HPHE pre-cooler with 118 heat pipes and designed for 30 °C ambient temperature can effectively dissipate 30 kW data center heat for most of the year. Size of the pre-cooler has strong dependence on the ambient temperature of the location.
- Heat Pipe Ice Storage System for 1 MW data center emergency cooling backup will require only ~ 68 heat pipes. 952 kg of ice per heat pipe per winter season can be generated. Cost of the heat pipe based ice storage system (0.3 to 0.68 \$/W) is comparable to the existing backup technologies (0.5 to 1 \$/W)
- Heat pipe based permafrost storage for agricultural products provides sustainable and environmental friendly approach for cold storages by saving electricity.
- Proposed heat pipe based cooling systems utilizes natural ambient cold energy and therefore there is no running cost and no greenhouse gas emissions
- Heat pipe heat exchanger with 65% effectiveness provided a waste heat recovery system that was able to supply all the heat needed by the proofing oven, thus eliminating the need for natural gas heating. On the basis of this experimental study, it was observed that the most suitable location for a gas reservoir to collect non-condensable gas in a loop type heat pipe heat exchanger is on the bottom header of the condenser where the condensate exits to the evaporator by gravity effect through adiabatic pipe.
- A medium-scale CPL with an aluminium evaporator, 31.70-mm outer diameter and 500-mm in length, and with UHMW polyethylene wick and acetone as the working fluid was designed and tested for car dashboard cooling. A minimum value of 0.19 °C/W for the total thermal resistance was achieved at the maximum heat load of 210 W with the corresponding evaporator temperature at 58.33 °C.
- Heat pipe turbine to convert available low grade heat into electricity was developed and tested. Sources. The prototype with cylinder of 2.8 m height and 0.5 m diameter was able to produce 100 W electrical power at 6000 rpm for 10 kW input heat load. Although the efficiency of the proposed heat pipe turbine is low, it has the capability of utilising very low grade heat and converts it into useful electricity.
- A demonstration system for geothermal heat extraction using large size heat pipe, 150 mm OD and 150 m long, with showering nozzles, for proper liquid distribution in the evaporator, was constructed and tested. The heat pipe was able to extract 90 kW heat continuously with 3000 W/m² heat flux at the evaporator.
- Combination of thermosyphon and thermoelectric cells provided a fully passive and simple power supply system for remote area applications using the temperature differences that exists in a typical solar pond. The designed system was able to provide maximum power point of 3.2 W which was obtained at 13.4 V and 0.24 A when the temperature difference of 27 °C was maintained cross 16 TEG modules. In this case, the open-circuit voltage and the short circuit current values of 26 V and 0.4 A respectively were obtained.
- Heat pipe provided an efficient cooling system for the trough-type solar concentrated photo voltaic module, with concentrator length 1 m, width 0.8 m, and aligned along east-west direction.
- Thermosyphon modules provided an effective heat transporting system for thermal management of the concentrated thermo electric generator module with paraffin wax phase change material thermal storage. Based on the simulation model, it was estimated that the designed system is able to maintain a

temperature difference of 152 °C across the thermo electric cells and produced 9.5W of thermoelectric power at concentration ratio of 75.

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REFERENCES

- Singh, R., Mochizuki, M., Mashiko, K., Nguyen, T., 2011, Heat Pipe Based Pre Cooler and Emergency Cooling Systems for Data Center Energy Conservation, Paper no. S051024, JSME 2011, Proceedings of the Japan Society of Mechanical Engineers Annual Conference, 11 – 14 September, 2011.
- Wu, X. P., Mochizuki, M., Mashiko, K., Nguyen, T., Nguyen, T., Wuttijumnong, V., Cabusao, G., Singh, R., Akbarzadeh, A., 2011, Cold Energy Storage Systems Using Heat Pipe Technology for Cooling Data Centers, *Frontiers in Heat Pipes*, **2**, 013005.
<http://dx.doi.org/10.5098/fhp.v2.1.3005>
- Wu, X. P., Mochizuki, M., Mashiko, K., Nguyen, T., Nguyen, T., Wuttijumnong, V., Cabsao, G., Singh, R., Akbarzadeh, A., 2010, Data Center Energy Conservation by Heat Pipe Cold Energy Storage System, IHTC-14, Proceedings of the International Heat Transfer Conference, Paper No: IHTC14-23128, 8-13 August, 2010, Washington DC, USA.
- Wu, X. P., Mochizuki, M., Mashiko, K., Nguyen, T., Wuttijumnong, V., Cabsao, G., Singh, R., Akbarzadeh, A., 2010, Energy Conservation Approach for Data Center Cooling Using Heat Pipe Based Cold Energy Storage System, 26th IEEE SEMI-THERM, Proceedings of the Semiconductor Thermal Measurement and Management Symposium, 21-25 February, 2010, Santa Clara, CA, USA, 115 – 122.
- Singh, R., Mochizuki, M., Mashiko, K., Nguyen, T., Wu, X. P., Wuttijumnong, V., 2011, Datacenter Backup Cooling System Using Heat Pipe Based Ice Storage, 48th NHTS, Proceedings of the 48th National Heat Transfer Symposium, 1-3 June, 2011, Okayama, Japan.
- Singh, R., Mochizuki, M., Mashiko, K., Nguyen, T., 2011, Heat Pipe Based Cold Energy Storage Systems for Data Center Energy Conservation, *Energy*, **36**, 2802-2811.
<http://dx.doi.org/10.1016/j.energy.2011.02.021>
- Tsuchiya, F., Ryokai, K., Mochizuchi, M., 1991, A Cold Storage of Agricultural Products Surrounded by Artificial Permafrost Using Heat Pipes, Proceedings of the 6th International Symposium on Ground Freezing, 10-12 September, 1991, Beijing, China, 537 – 542.
- Mashiko, K., Ryokai, K., Tsuchiya, F., Fukuda, M., 1989, Development of an Artificial Permafrost Storage, Proceedings of the ASME/AIChE National Heat Transfer Conference, 5-8 August, 1989, Philadelphia, Pennsylvania.
- Dube, V., Akbarzadeh, A., Andrews, J., 2004, The Effects of Non-Condensable Gases on the Performance of Loop Thermosyphon Heat Exchangers, *Applied Thermal Engineering*, **24**, 2439–2451.
<http://dx.doi.org/10.1016/j.applthermaleng.2004.02.013>
- Singh, R, Akbarzadeh, A, Mochizuki, M, 2004, Capillary Pumped Loops: An Overview, Proceedings of the 1st International Seminar on Heat Pipes and Heat Recovery Systems, 8-9 December, 2004, Kuala Lumpur, Malaysia.

Singh, R., Akbarzadeh, A., Dixon, C., Riehl, R. R., Mochizuki, M., 2004, Design, Analysis and Investigation of a Medium Scale Capillary Pumped Loop, Proceedings of the 13th International Heat Pipe Conference, 21-25 September, 2004, Shanghai China, 253-261.

Singh, R., Akbarzadeh, A., Mochizuki, M., 2004, Effective Thermal Management Using Capillary Pumped loop, Proceedings of 1st International Forum on Heat Transfer, Paper No: GS2-23, 24-26 November, 2004, Kyoto, Japan, 115-116.

Singh, R., Akbarzadeh, A., Mochizuki, M., 2008, Thermal Performance of a Capillary Pumped Loop for Automotive Cooling, Experimental Heat Transfer, **21** (4), 296 — 313.

<http://dx.doi.org/10.1080/08916150802291129>

Nguyen, T., Mochizuki, M., Mashiko, K., Sauciuc, I., Akbarzadeh, A., Johnson, P., Kusaba, S., Suzuki, H., 1999, Heat Pipe Turbine for Production of Electrical Power from Renewable Sources, Proceedings of the 5th ASME/JSME Joint Thermal Engineering Conference, ASME Paper No: AJTE99-6339, 15-19 March, 1999, San Diego, California.

Mashiko, K., Mochizuki, M., Watanabe, Y., Kanai, Y., Eguchi, K., Shiraishi, M., 1994, Development of a Large Scale Loop Type Gravity Assisted Heat Pipe Having Showering Nozzles, Proceedings of 4th International Heat Pipe Symposium, 16-18 May, 1994, Tsukuba, Japan, 264-274.

Mochizuki, M., Mashiko, K., Shihyakugari, S., Watanabe, Y., Eguchi, K., Namiki, T., Shiraishi, M., 1995, A Demonstration Test of a Large Scale Loop Type Heat Pipe With Showering Nozzles, Proceedings of the 9th international heat pipe conference, 1-5 May, 1995, Albuquerque, USA.

Akbarzadeh, A., Andrews, J., and Golding, P., 2005. Solar Pond Technologies: A Review and Future Directions, Chapter 7, Advances in Solar Energy, Earthscan.

Dunn, P. D., Reay, D. A., 1994. Heat Pipes, Pergamon, Fourth Edition.

Rowe, D. M., 1995, CRC Handbook of Thermoelectrics, CRC Press.

Tundee, S., Singh, R., Terdtoon, P., Akbarzadeh, A., 2008, Power Generation from Solar Pond Using Combined Thermosyphon and Thermo Electric Modules, Proceedings of the SET2008 - 7th International Conference on Sustainable Energy Technologies, 24-27 August, 2008, Seoul, Korea.

Singh, R., Tundee S., Akbarzadeh, A., 2011, Electric power generation from solar pond using Combined Thermosyphon and Thermoelectric Modules, Solar Energy, **85** (2), 371–378.

<http://dx.doi.org/10.1016/j.solener.2010.11.012>

Akbarzadeh, A., Wadowski, T., 1996, Heat Pipe Based Cooling Systems for Photovoltaic Cells under Concentrated Solar Radiation, Applied Thermal Engineering, **16** (1), 81-87.

[http://dx.doi.org/10.1016/1359-4311\(95\)00012-3](http://dx.doi.org/10.1016/1359-4311(95)00012-3)

Tan, L. P., Date, A., Singh, R., Akbarzedah, A., 2010, Passive Cooling of Concentrated Photovoltaic Cells Using Phase Change Material, Solar2010, Proceedings of the 48th AuSES Annual Conference, 1-3 December 2010, Canberra, Australia.

Tan, L. P., Singh, R., Date, A., Akbarzadeh, A., 2011, Thermal Performance of Two Phase Closed Thermosyphon in Application of Concentrated Thermoelectric Power Generator using Phase Change Material Thermal Storage, 10th IHPS, Proceedings of the 10th International Heat Pipe Symposium, 6-9 November, Taipei, Taiwan.