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THE EFFECT OF SINK TEMPERATURE ON A CAPILLARY PUMPED LOOP EMPLOYING A FLAT EVAPORATOR AND SHELL AND TUBE CONDENSER

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ABSTRACT

An experimental facility for conducting research on capillary pumped loop (CPL) systems was developed. In order to simulate shipboard cooling water encountered at various locations of the ocean, the heat sink temperature of the facility could be varied. A flat plate, CPL evaporator was designed and tested under various heat sink temperatures. The sink temperature ranged from 274.3 to 305.2 K and the heat input varied from 250 to 800 W which corresponds to heat fluxes up to 1.8 W/cm^2 . The CPL flat plate evaporator performed very well under this range of heat input and sink temperatures. The main result obtained showed that a large degree of subcooling developed between the evaporator vapor outlet line and liquid return line. This condensate depression increased with increasing heat input.

INTRODUCTION

Capillary Pumped Loops, CPL, are systems that employ the evaporation and condensation processes of a working fluid to transport heat. They are related to heat pipes except for two significant differences: the CPL components are separated and the CPL system typically employs a liquid reservoir. A simple schematic for a CPL system is depicted in Figure 1. In this scheme the CPL system is used to remove the heat generated from onboard ship electronic components and to transfer that heat to a shell and tube condenser where it can be removed by sea water. The major components in this simple design are the evaporator, condenser, reservoir, vapor and liquid lines.

Capillary pumped loops were first proposed in the USA by Stenger [1966]. They have been primarily used for space systems where the condenser/radiator was typically a cold plate on which condenser tubes were mounted.

Figure 1 - CPL Schematic

An experimental facility has been designed and constructed at the U S Naval Academy to study the shipboard use of CPL systems. This facility employs a typical shell and tube condenser. The reported investigation is the first study for the facility. In this investigation, a flat plate CPL evaporator was fabricated. Since Navy ships travel through waters of varying temperatures, this study investigates the effect of varying sink temperature, between 274.3 and 305.2 K, on CPL performance under a heat load of 250 to 800 W. Traditionally, CPL evaporators have been of cylindrical geometry, Faghri [1995]. The cylindrical design does not readily lend itself to the mounting of flat electronic components, hence, the flat plate evaporator was built.

Yan and Ochterbeck [2000] proposed a two-dimensional, steady state numerical study for a cylindrical geometry that was based on a two-phase mixture model by Wang and Beckermann [1993]. They derived solutions for the heat transfer in the liquid core and wick structure of the CPL evaporator. The effects of heat load, liquid subcooling and wick thermal conductivity on liquid and wick temperatures were studied. Results showed that increasing the heat load and liquid subcooling decreased the temperature in the liquid core. The increasing heat load increased the working fluid velocity which enhanced convection and the increase in subcooling decreased the length of the two-phase region.

Kiper et al. [1990] studied a CPL system under transient mode operation. They employed step power inputs to the evaporator. They concluded that initial inlet subcooling was an important parameter for successful CPL operation. Their evaporators would dry out under low liquid subcooling conditions.

Ku et al. [1993] designed a CPL system with a see-through evaporator so that the inside flow of the working fluid could be visualized. They reported that inlet subcooling was also found to be very important. Start-up was always successful when sufficient inlet subcooling was provided. Under high heat loads, the investigators reported that vapor bubbles would form in the wick structure and the evaporator would ultimately de-prime.

Li et al. [1999] also reported start-up visualization studies with polyethylene wicks. Under load sharing conditions, the authors reported that the liquid region was subject to pressure oscillations. The oscillations were believed to be caused by the expansion and subsequent collapse of vapor plugs in the wicking structure. Li and Ochterbeck [1999] reported that thick wicks with low thermal conductivity exhibited, at start-up, large liquid subcooling with increasing heat input.

EXPERIMENTAL FACILITY

The experimental facility consisted of two main loops, a chiller/heater loop and a CPL evaporator/condenser loop. The chiller/heater loop was used as the thermal regulation loop for the CPL condenser. A schematic of the chiller/heater loop is depicted in Figure 2. Essentially, a water/ethylene glycol bath was fitted with a dual temperature control regulation system. The two temperature control system was designed to regulate the bath temperature within a desired temperature bandwidth either above or below the ambient temperature of the laboratory room. Two, 3 kW immersion heaters mounted in the water/glycol tank were utilized to maintain CPL sink temperatures to 32.2 C in this study. A 25 GPM centrifugal pump took suction from the bath and discharged the water/glycol mixture through a chiller barrel and back into the bath. The bath could also use water if the desired CPL system sink temperature was always above 0 C.

Figure 2 - Thermal Control Loop

The cold side of the chiller barrel was connected to an R134a/404a, 3 ton, 5 horsepower, water cooled compressor unit. The compressor was capable of handling either R134a or R404a. The selection of the refrigerant depended upon the desired CPL sink temperature. R134a was used in this study since the sink temperature was not allowed to drop below 0 C. R404a could be used to test CPL working fluids to a temperature of about -40 C. The compressor was equipped with a 3 ton thermal expansion valve and a hot gas bypass system. The condenser for the compressor unit was water cooled. In this study, the water/glycol bath was maintained between 1.1 and 32.2 C with a 2 degree bandwidth for compressor/heater cycling. Due to the relatively low heat inputs used by the CPL evaporator, this system worked very well. When the compressor unit was in operation, the temperature drop of the refrigerant across the chiller barrel was less than 1.5 C.

The bath temperature was monitored with type-K thermocouples and two Johnson controller units, one for the heaters and bath temperatures above ambient, the other for the chiller and bath temperatures below ambient.

A schematic of the CPL evaporator/condenser loop is shown in Figure 3. The water/glycol mixture in the bath was drawn by a 50 GPM centrifugal pump and discharged through four flowmeters. The pump capacity far exceeded the combined capacity of the flowmeters so ball valves were placed upstream of each flowmeter to regulate the flow to 100 percent of scale. This corresponded to a maximum flow of 6.5 GPM per flowmeter bringing the total coolant flow to 26 GPM. The water/glycol mixture was passed through the CPL condenser and returned back to the bath in the chiller loop. Type-K thermocouples were placed in the inlet and outlet lines of the four cooling tubes to the CPL condenser. The typical temperature drop along a cooling tube was less than 0.1 C for the full 26 GPM of the coolant flow. In this study, the pump

was operated at maximum flow conditions in order to keep the tube temperature drops low.

The capillary pump loop consisted of a CPL evaporator connected to the CPL condenser by means of a vapor header and vapor tube. The vapor generated by the CPL evaporator entered the side of the CPL condenser and condensed on the water/glycol cooling tubes. The CPL condenser was fabricated from a 0.33 m diameter, 0.67 m long, 0.0095 m thick SS304 shell. The end plates were fabricated from SS304 plate. The 0.0191 m OD type L copper condenser tubes were arranged in a 0.05 m pitch diamond pattern to prevent the tubes from dripping onto each other and were held in place with Swage-Lok fittings. The condenser hot well was fabricated from 0.0762 m diameter, SS304 Schedule 40 pipe. Four view ports made from SS304, 0.1016 m OD Schedule 40 pipe were fabricated, two on each side, into the sides of the CPL condenser, so that the vapor condensation process could be monitored to study the condensation heat transfer effects of different types of cooling tubes. The condenser shell had three penetrations for type-K thermocouples to monitor the temperature inside. Two of these penetrations were in the upper part of the shell, and one was in the lower portion. Three other penetrations were made axially along the top of the condenser. These penetrations were used for a vacuum pump/charging line, a mechanical pressure gauge and an electronic pressure transducer.

Figure 3 - CPL Evaporator/Condenser Loop

Condensate from the hot well flowed through a 0.95 cm OD stainless steel tubing, needle valve, 0.95 cm OD liquid distribution header and to the three 0.95 cm OD tube entrances to the wicking structure of the CPL evaporator. The working fluid was distilled water. A differential pressure transducer was placed across the needle valve so that the liquid return line flow resistance could be measured when increased to simulate various effects such as increased return line pipe length, or gravity head differences brought about by elevation changes

between the evaporator and the condenser. Upstream of the needle valve and downstream of the condenser hot well a tee was placed which tied into the liquid reservoir. The liquid reservoir was fabricated from a 1 liter stainless steel bottle and a stainless steel valve. The liquid reservoir line temperature was also monitored after the valve with a type-K thermocouple. Figure 4 shows a photograph of the CPL condenser.

Figure 4 - CPL Condenser

FABRICATION OF CPL EVAPORATOR

The CPL evaporator incorporated a flat plate design as depicted in Figure 5. The evaporator was fabricated from two brass plates 1.77 cm thick by 35.56 cm long by 25.4 cm wide. A 0.635 cm, 15.24 by 30.48 cm recess was milled out in the bottom brass plate to accommodate the wicking material and the vapor grooves milled into the top brass plate. In the bottom recess, three holes, each of 0.953 cm in diameter were bored out to provide access to the evaporator from the liquid return line. The holes were connected to the liquid return line with 0.953 cm OD copper tubing. To seal the top and bottom halves of the evaporator, a groove was machined around the 15.24 cm by 30.48 cm recess to accommodate a 0.635 cm O-ring.

The CPL evaporator was designed to be heated from the top side. The top plate of the evaporator was fabricated from a 1.77 by 35.56 by 25.4 cm piece of brass. A 0.643 cm deep border, 5.08 cm wide along the underside of each long side and 2.54 cm wide along each short side was milled into the plate. Sixty-four grooves of dimensions 0.238 cm wide, 0.635 cm deep, and 15.24 cm long were machined into the underside of the top brass plate along the width. These grooves formed the vapor channels. The extra 0.0076 cm milling of the border on the top brass plate assured that the fins would slightly

compress the wick positioned in the bottom brass plate. On one end of the grooves, 1.77 cm was taken off of every groove to form an internal vapor manifold. In this 1.77 cm wide by 0.064 cm deep channel, three 0.953 cm holes were bored and connected to the 1.77 cm OD external vapor header by means of 0.953 cm OD copper tubing.

Figure 5 Schematic of CPL Evaporator

These grooves left a series of fins and channels in the top brass plate. The fins delivered thermal energy from the heat source to the wicking material, and the channels were the vapor paths for the evaporated working fluid.

The wicking material was made from a 0.318 cm thick, 15.24 by 30.48 cm, porous, synthetic, high density, polyethylene material obtained from the Porex Corporation. The wick had an effective pore radius of 20 micrometers, a permeability of $4.65 \times 10^{-12} \text{ m}^2$, a thermal conductivity of 60.5 W/m-K and a porosity of 50%. A relatively thin wick was chosen to minimize the pressure drop across the wick. Polyethylene was chosen to act as a thermal barrier between the saturated vapor in the grooves and the subcooled water from the liquid side of the evaporator plate.

The wick, and two brass plates were bolted together through ten 0.635 cm holes in the border region to form the evaporator plate. Figure 6 depicts the final CPL evaporator assembly.

The top surface of the CPL evaporator was heated by means of four 7.62 by 15.24 cm Kapton thin film heaters. Each heater was rated at 208 V and 900 W. Two heaters were electrically connected in parallel to form a 208 V, 1.8 kW heater. The two, 1.8 kW heater assemblies were individually controlled by voltage transformers.

Measurement instruments for the CPL evaporator plate included 18 type-K thermocouples placed in three rows of six thermocouples separated by a pitch of 5.08 cm. The thin film heater strips were then placed on top of the thermocouples. A 15.24 by 30.48 by 0.635 cm thick Bakelite slab was placed over the heater strips to insulate the back side of the thin film heaters, and a 0.635 cm thick, 15.24 by 30.48 cm piece of brass was placed over the Bakelite. Type-K thermocouples were also placed on the center of the brass bottom plate, and

inside the liquid return line and outgoing vapor line close to the CPL evaporator. A differential pressure transducer was placed across the liquid return and outgoing vapor lines to monitor the pumping head developed by the wick.

Figure 6 - CPL Evaporator Assembly

EXPERIMENTAL PROCEDURE

The CPL system was evacuated with an oil diffusion vacuum pump running for several hours. Through preliminary vacuum tests at ambient temperature, it was determined that there was a slight leak through the CPL evaporator O-ring seal. Air was bleeding in at a rate of 6.9 kPa every 3 hours. At CPL operating conditions, the air leakage rate would be smaller due to the lower pressure differential between ambient room and the evaporator internal pressures.

The CPL loop was charged with distilled water through a water column and ball valve mounted on the top of the stainless steel reservoir. The ideal working fluid charge (100% charge) was calculated by assuming that the condenser hot well, liquid return line, and evaporator wick was filled with water, and the condenser shell, vapor lines and vapor grooves were filled with vapor. This ideal charge was 0.334 liter. Due to the comparatively large condenser and reservoir, it was decided to use a charge 300% of ideal or 1 liter of water. All tests were conducted with 1 liter of distilled water.

For the conducted experiments, the standard position for the isolation valve between the two-phase reservoir and the CPL system was fully open. In this configuration, the reservoir served as a depository for excess working fluid. The needle valve, which could be used to increase the pressure drop in the liquid return line, was also wide open. CPL steady state was defined as the point when the system temperature

approached a constant value. This could be most readily seen by the heated surface thermocouples on the CPL evaporator.

A range of power inputs, 0 to 800 W, were tested at three sink temperatures, 1.1, 18.1 and 32.2 C, respectively. The temperatures were chosen to bracket the temperatures of ocean water that Navy ships would be exposed to. The following is a typical procedure for operation at 18.1 C:

1. Set the sink temperature to 18.1 C and allow the chiller loop to attain steady state through the CPL condenser.
2. Start the data acquisition system.
3. Increase power input to 250 W. Record data and wait for steady state to approach. The CPL loop initiated start-up and excess working fluid was driven out of the evaporator.
4. Increase power level by approximately 50 W, await steady state.
5. Repeat (4) until 800 W.
6. Shut down system.
7. Allow system to cool overnight.
8. Repeat procedure for 1.1 and 32.2 C.

Another project goal was to test the ability of the CPL system to respond to a varying sink temperature. In two experiments, the sink was allowed to rise in temperature from 1.1 to 32.2 C and to drop in temperature from 32.2 to 1.1 C. The procedure for these tests was as follows:

1. Set the sink temperature to 1.1 C and perform CPL start up.
2. Wait for steady state.
3. Set power level to 550 W.
4. Increase sink temperature in increments to 32.2 C.
5. Wait for steady state at the final temperature.

The computer controlled data acquisition system monitored all thermocouple data, absolute and differential pressure electronic transducer data, and the power input to the heating surface. The measurement system data collection frequency was 0.1 Hz. Since some regions of the brass capillary pumped loop evaporator, such as the unheated bottom surface, were exposed to the ambient environment, an estimation of the heat losses or gains from the input power had to be made in order to correctly calculate the heat input to the working fluid. The losses/gains depended on the heat input setting. A convective-conductive resistance type analysis showed that the maximum heat loss/gain based on actual temperatures collected, ranged from 4 to 9% of the input power. Figures 12 and 13 show the corrected heat input levels.

Finally, the condenser shell, liquid return and vapor lines were left un-insulated to further simulate a possible shipboard environment. The subcooling introduced by the un-insulated liquid return and vapor lines was less than 3 C and 1 C, respectively. The bulk of the subcooling occurred in the condenser. As mentioned in the papers reviewed in the introduction, subcooling does help prevent wick dry out and

consequent CPL system failure. For the range of heat inputs tested, 250-800 W, no system failure was observed.

RESULTS AND DISCUSSION

In the following series of plots, the nomenclature used is:

Pump IN – entering liquid temperature.

Pump OUT – exiting vapor temperature.

Plate Bottom – temperature of the bottom evaporator surface.

AVG Surface- average temperature of the heated evaporator surface.

Sink Temp – temperature of the heat sink bath.

A. Constant Heat Sink Temperature

Figure 7 displays the temperature-time history for the CPL system operating at a heat sink temperature of 1.1 C. The initial system heat input was 270 W. The changes in power input can be readily seen by the curve representing the average top surface temperature of the evaporator plate. The thermocouples were located between the heaters and the top surface, so the temperature represented is indicative of some average temperature between the evaporator surface and the heaters. The duration of the first heat input level (270 W) is shown to be slightly longer than the duration of the subsequent heat input levels. This is due to the fact that at start-up, the heat transfer processes are starting to develop, hence they have smaller rates than at full operating conditions.

The liquid inlet temperature to the evaporator is shown by the bottom line. At first, during start-up, this temperature is seen to dip slightly below the ambient room temperature of about 292 K. This is due to the cold cooling water temperature in the condenser tubes. The evaporator inlet temperature increased very slightly over the duration of this experiment. The temperature that is most representative of the operating temperature in the evaporator grooves (vapor side of the wick) is the vapor outlet temperature. As can be seen, the changes in power input level cannot be discerned on this temperature curve. The curve shows a very smooth monotonic increase in temperature with increasing input power. The reason the evaporator pump outlet temperature appears monotonic rather than exhibit the discrete changes in temperature that the average plate surface temperature exhibits is due to the large heat capacity (thickness) of the brass evaporator cover plate. The temperature at the bottom surface of the evaporator shows subcooling. This subcooling increases with increasing heat input.

The temperature of the vapor outlet is assumed to be at the saturation temperature. The subcooling or condensate depression, defined as the saturation temperature of the outgoing vapor minus the entering liquid temperature, is approximately 50 C at a power level of 800 W.

Figures 8 and 9 show the temperature-time histories for the experiments run at heat sink temperatures of 18.1 and 32.2 C, respectively. The data shows the same general trends as the data in Figure 7. One slight difference is at start-up. The

entering evaporator liquid temperature is not seen to drop below the ambient room temperature because the heat sink temperatures are above room temperature. For the last heat input case of approximately 780 W, the degree of subcooling between the vapor outlet temperature and entering liquid temperature was approximately 32 and 35 K for the heat sink cases of 18.1 and 32.2 C, respectively.

The results of the data taken at the three heat sink temperatures are displayed on Figure 10 as the effect of heat input on system subcooling. The largest condensate depression as a function of heat input is for the case of the lowest heat sink temperature, 1.1 C.

Figure 11 shows a typical pressure history for the 18.1 C sink case. The pressures portrayed are the differential pressures across the evaporator (from vapor outlet to liquid inlet), and across the needle valve between the condenser hot well and the CPL evaporator. For the duration of these tests, this needle valve was wide open and no concern to this study.

The differential pressure across the evaporator pump was defined as the pressure of the vapor outlet header minus the pressure of the liquid return header. Since the vapor outlet header was 3.81 cm above the evaporator plate, and the liquid return header was 3.81 cm below the evaporator plate, initially, there existed a negative differential pressure due to the hydrostatic head of the working fluid. This is shown in Figure 11. As the CPL evaporator heats up and primes, water in the outgoing vapor paths is replaced by vapor and the differential pressure decreases. Once the liquid-vapor interface reaches the wick, a positive pressure due to the capillarity of the wick is achieved. The maximum pumping pressure differential of this CPL evaporator, taken at a heat flux of 780 W and correcting for the hydrostatic head on the liquid side of the evaporator was approximately 0.35 kPa.

Figure 7 - Constant heat sink, 1.1 C

Figure 8 – Constant Heat Sink, 18.1 C

Figure 9 – Constant Heat Sink, 32.2 C

Figure 10 The Effect of Heat Input on Subcooling

Plots of differential pressure for the other heat sink cases produced similar results. One note, the heat sink temperature of 274.3 K, the capillary pumping pressure across the wick was negligible. This seemed to indicate that the system operated via the hydrostatic head in the liquid return line much like a thermosyphon.

B. Varying Heat Sink Temperature

Two experiments were conducted with a constant heat input of approximately 550 W, but the sink temperature was allowed to vary from 18.1 to 32.2 C over a 2 hour and 20 minute period, and visa versa.

The results for the increasing heat sink temperature experiment are depicted in Figure 12. The temperature of the heat sink is also displayed on this figure. The data shows that the CPL system is capable of smoothly adjusting through the selected ranges of heat sink temperatures. One interesting note on Figure 12 is that when the sink temperature starts out very cold, the system appears to head towards a large condensate depression across the evaporator. As the sink temperature warms up, this condensate depression decreases.

The results for the decreasing heat sink temperature experiment at 550 W are shown in Figure 13. Initially, the heat sink temperature is shown to be higher than the starting liquid return line and outgoing vapor line temperatures of the evaporator. As long as this condition exists, the CPL condenser will not cool the system and the result will be a heating up of the system until the evaporator temperatures are higher than the sink temperature. This is seen by the very steep initial average temperature of the top, heated surface. Once the sink temperature falls below the evaporator temperature, cooling takes place.

Figure 11 – Pressure History

Figure 12 Increasing Sink Temperature Transient

Figure 13 - Decreasing Sink Temperature Transient

CONCLUSIONS

An experimental facility for conducting research on capillary pumped loop systems was developed. The facility has the ability to vary the heat sink temperature in order to simulate shipboard cooling water encountered at various locations of the ocean. The heat sink temperature ranged from 1.1 to 32.2 C and the heat input varied from 250 to 800 W which corresponds to heat fluxes up to 1.8 W/cm². The following can be concluded:

- The CPL system transitioned very smoothly with power level changes at all sink temperatures.
- There was a degree of subcooling between the evaporator vapor outlet and liquid inlet lines. This degree of subcooling increased monotonically with increasing heat input.
- At low sink temperatures, the degree of subcooling was on the order of 50 K.
- The maximum capillary pumping head of the polyethylene wick during these experiments was achieved at a heat input of 780-800 W, with a sink temperature of 18.1 C, and was 0.35 kPa.
- For the cases of varying sink temperatures at a constant 550 W of heat input, the CPL system was shown to respond very smoothly.

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