Experimental study of a capillary pumped loop in comparison with the prediction of a 3D CFD

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Abstract
A comparison is presented between experimental and numerical results regarding the operation of a capillary pumped loop evaporator. Two cylindrical evaporators were tested, with different heated porous lengths, 20 and 40 mm, respectively. Both have 22 mm external diameter, 9 mm porous thickness and 80 mm porous length. The working liquid was water. The loop was made from copper tubes and the evaporator from copper porous wick covered with aluminum with grooves formed in the inner surface. All tests took place on a horizontal level using heat load applied to the evaporator surface from an 85-W electric resistance. The experimental measurements were compared with the predictions of a three-dimensional CFD model of the evaporator and were found to be in satisfactory agreement. For the 20-mm wick heated length evaporator CFD model with water initial temperature of 20°C the divergence with the experimental pressure drop mean value was 0 Pa for volume flow rates between 0.4 and 0.6 l/min and 50 Pa for the rest of the values. For 30°C the divergence was 0 Pa <0.4 l/min and 50 Pa for larger flow rates. Moreover, for 40°C the difference was up to 50 Pa from 0 to 0.9 l/min. In every case predictions were below the wick capillary limit. The computed outflow temperature presented a maximum difference of 1.5% compared with the experimental data, which is very satisfying. On the other hand, the predictions of the evaporator CFD model with a 40-mm wick heated length were even better.

Keywords: capillary pumped loop; three-dimensional model; thermal resistance; heat transfer coefficient

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Received 27 March 2013; revised 10 April 2014; accepted 25 July 2014

1 INTRODUCTION

The capillary pumped loop (CPL) is considered to be a reliable two-phase thermal management device. It operates passively by means of capillary forces generated on a porous structure present in the evaporation section, which are responsible for driving a working fluid from a high-temperature source to a low-temperature sink. That means there is no need for mechanical pump for driving the fluid. A typical CPL system consists of a capillary evaporator (heat acquisition section), a condenser (heat dissipation section), a two-phase reservoir, liquid and vapor lines.

Capillary forces are generated in the capillary evaporator, which acquires heat from an external heat source and transfers it to the working fluid. Vapor formation inside the wick is responsible for liquid displacement in the vapor line toward the condenser during the start up. A reservoir is used to set the temperature at which the entire loop will operate. The CPL works without moving parts and very little power consumption, thus making it a reliable solution.

It was suggested that some features of the surface encourage bubble nucleation. It has been observed that nucleation occurs in cavities within the surface; these cavities contain minute bubbles of trapped gas or vapor which act as the starting points for bubble growth. When the bubble leaves the site, a small bubble remains in the cavity which acts as the start for the next bubble [1].

Theory states that the CPL fluid flow behaves in the same manner as a simple spring-mass-damper dynamical system...
susceptible to the modulation of external disturbances. Under certain system conditions an infinitesimal disturbance entering the CPL acquires energy from the system and grows in magnitude to a point that it completely changes the equilibrium state of the system—causing instability. The amplification factor of the pressure oscillation is a function of the CPL system design and its operating conditions. Hence, the pressure oscillation is an inherent characteristic of a CPL.

The first CPL was developed by Stenger [2] at the end of 1960s where low gravity experiments were presented on board STS shuttles, while Europeans tested their own device in the early 1990s so that new prototypes would be tested during the flight of Stentor satellite [3]. These developments aimed at increasing the dissipated powers in telecommunication satellites [4] and generally to launch such devices as state-of-the-art technological designs for thermal control of future spacecrafts. Today, such devices are tested to provide thermal control in satellites such as the Earth Observing System, the Hubble Space Telescope and the Mars Surveyor [5]. Other trends are the CPL for solar heating applications, where the evaporator was connected to a flat plate collector and the working fluid was either acetone or ammonia. The purpose was to investigate the possibility of CPL being an alternative to residual and commercial water heating systems [6]. In the late 2000s it was illustrated the potential of the CPL as an effective temperature control device in automotive applications [7]. Finally, in 2012 CPL, as part of a cooling system, was proposed as an alternative for the cooling and thermal control of a proton exchange membrane fuel cell (PEMFC) [8].

A first approach to CPL operation was the development of an experimental set-up to investigate the performance of a loop heat pipe as capillary pump, as shown by Diamantis et al. [9]. The work was performed at the Laboratory of Fluid Mechanics and Energy (LFME), University of Patras, Greece, as part of an EC-funded research project, entitled TRI-GEN EGD, which aimed at the development of a novel Tri-generation Electrogaodynamic converter system. The capillary pump investigated was used to pump the working liquid of the system, i.e. water, using external waste heat. From the experimental results, a strong potential of the sintered material filled pipes to work as capillary pumps was clear. When the system was operated under steady-state conditions there was a constant periodic flow at an average value of 2 l/h.

These experimental results were satisfactorily predicted by a theoretical model of the CPL developed by Margaris et al. [10]. The theoretical model requires that the overall pressure drop in the loop should be less than the maximum capillary pressure in order to ensure that the system will operate continuously. The comparison between experimental and theoretical results, concerning the pressure head rise vs. flow rate, showed a very good agreement for high flow rates, while for lower flow rates the agreement was poor because of the unsteady nature of flow at the start-up period.

The unsteady nature of flow was also observed/studied experimentally in a CPL, where the system displayed similar behavior [11]. Differential pressure transducers measuring the pressure drop across the evaporator in the CPL, in order to characterize the system performance, displayed the oscillatory behavior of the differential pressure. Through flow visualization within the capillary evaporator, the pressure oscillations were observed to correlate with a radial liquid motion in and out of the wick.

It is evident that the pressure oscillations are the result of certain instabilities in the CPL system. Pressure oscillation will be stable if the energy content of the instabilities is smaller than the CPL energy dissipation. Otherwise, this becomes unstable. The frequency and amplitude of the pressure oscillations are functions of the physical dimensions of the CPL components, such as transport line size, vapor volumes and operating conditions, such as power input and condenser sink temperature.

The effect of the size of the transport lines on the pressure oscillations was studied through a comparison between the predictions of a hydrodynamic model, developed for predicting the pressure oscillations, and experimental data, when liquid and vapor lines are changed during tests [12]. Reducing the liquid line diameter had a severe effect on the amplification of the pressure oscillations, while decreasing the vapor line diameter had a minor effect on the system pressure oscillations.

The heat transport capability of a CPL was also limited by the pressure drop that a wick could sustain. As mentioned, the pressure drop is not constant, even under seemingly steady operation. The just mentioned hydrodynamic model was also employed to predict/study the pressure oscillations of a CPL from another series of experiments [13]. In order to verify the effect of various parameters on the pressure oscillations, experiments were conducted varying the evaporator design, the power input, the condenser sink temperature and flow resistance between the reservoir and the loop. The predictions of the hydrodynamic model were shown to be in excellent agreement with the experimental measurements.

The key conclusion of the just-described research was that when the peak of the pressure drop oscillation exceeds the capillary limit, given in Equation (1) as a function of the surface tension, σL, and the porous wick radius, rc, of the wick, under any operating conditions, the vapor could penetrate the largest pores of the wick creating an intermittent injection of vapor from the pump vapor grooves into the liquid core (Figure 1).

\[ \Delta p_{c,max} = \frac{2\sigma_L}{r_c} \] 

(1)

If the return liquid was sufficiently subcooled, the vapor bubble was prevented from growing, therefore allowing the capillary pump to operate. When the subcooling of the return liquid was reduced, the vapor bubble expanded toward the pump inlet which blocked off the liquid supply and deprived the wick. The system pressure drops during the normal operation includes the pressure drop in the evaporator, \( \Delta p_w \), in the liquid and vapor line, \( \Delta p_L \) and \( \Delta p_v \), respectively, the condenser, \( \Delta p_C \), and due to gravity, \( \Delta p_g \), as given in the following equation [10]:

\[ \Delta p_{total} = \Delta p_w + \Delta p_L + \Delta p_v + \Delta p_C + \Delta p_g \] 

(2)
In the present article, the predictions of a three-dimensional CFD model of the evaporator are compared with the experimental measurements of a CPL evaporator. First, the three-dimensional CFD model is presented. Secondly, the experimental CPL system and the experiments performed are described. Next, the predictions of the three-dimensional CFD model of the evaporator are compared with the experimental measurements of the capillary pump loop evaporator. Finally, the conclusions of the work performed are presented.

2 CPL SYSTEM DESCRIPTION AND EXPERIMENTS PERFORMED

The experimental set-up diagram is presented in Figure 2 with its main components numbered. The working liquid heated tank (1) with a 1500-W resistance and a thermostat to control working liquid temperature. The liquid line (2) driving water from the tank to the evaporator inlet, the evaporator (3) containing the porous wick for vapor formation, the electric resistance (4) to apply the necessary heat load to the evaporator external wall, the vapor line (5) driving two-phase flow from the evaporator outlet to the condenser, the condenser (6) to cool down the working liquid using a fan to supply ambient air around the copper tube. A vacuum pump (7) was used to remove any remaining from the loop before every experiment. Data acquisition system was composed of temperature sensors (8, 9) to measure temperature at the evaporator inlet and outlet, pressure sensors (10, 11) to measure pressure at the evaporator inlet and outlet, temperature sensors at the condenser (12, 13), a volume flow rate meter (14) and valves (15–22) to control the fill-up procedure of the capillary pump loop.

Figure 3 is a picture of the experimental configuration. Both liquid and vapor lines were made of copper tube with a diameter of 18 mm covered with proper insulation in order to reduce heat losses to the environment. The electric resistance around the evaporator was 80 W. The temperature sensors were of the model TN2530 with a 0.2-K accuracy, the pressure sensors were of the model PN3004 of less than $+0.5$ bar accuracy and the volume flow sensor was of the model SU7000 of less than $\pm 3\,\text{MW} + 0.2\%$ MEW accuracy for water, all from IFM Electronics Gmbh, connected to a data acquisition system on a PC. The data acquisition system card was a National Instrument Fp-Ai-110 with eight 4–20-mA inputs.

The first of the two evaporators used in the loop is shown in Figure 4. It had a 22-mm diameter and 100-mm length.
containing a porous wick of 9 mm thickness and 90 mm length made from copper. The evaporator inlet and outlet had a diameter of 4 mm. The porous wick was covered with aluminum of 9 mm thickness with four grooves, 2 mm high and 6 mm wide, formed in the inner surface guiding vapor from the porous wick to the outlet. The porous heated length, \( L_{xf} \), was 20 mm of 80 mm total porous length. The second evaporator had a porous heated length, \( L_{xf} \), of 40 mm of 80 mm total porous length.

Both evaporators were tested under the same conditions in order to establish a comparison basis. The loop was filled with water in three different temperatures, 293, 303 and 313 K. The initial loop pressure was regulated at the zero level in order to make it easier to measure the loop pressure difference created by the evaporator between the inlet and outlet.

By keeping the pressure as low as possible inside the loop, the thermophysical properties of water for smaller saturation pressure were taken advantage of. Accordingly, the liquid was vaporized by applying smaller heat load at the external wall of the evaporator. Normally, for saturation pressure of 1 bar, water would boil at 373 K, requiring 2251 kJ/kg. In the present case, the significant smaller initial loop pressure could result in smaller water saturation pressure, in other words, less heat load is needed to vaporize the amount of working liquid inside the evaporator wick.

Table 1 summarizes the experimental operating conditions to be used also in the CFD model. They were divided into two sets, according to the heated length of the wick, with three sub-cases, for each initial temperature of the working fluid. The saturation temperature was not measured directly, but was derived from the pressure of the loop; in each case the evaporator was under steady operating state. For the remainder of the article, each case studied will be referred to as stated in Table 1, e.g. Set-1a for 20 mm heated wick, 293 K initial temperature and 323 K saturation temperature.

Considering that each experiment lasted 24 h, a short, representative time window of the data acquired is shown. Figure 5 shows an image of the pressure oscillations at the evaporator inlet and outlet for Set-1a experiment under steady-state operating condition of the CLP for a short period of 600 s. Each curve stands for the pressure at evaporator inlet and outlet, respectively, showing the rapid change of pressure, mostly at the inlet. At this point we should remind that charging pressure before applying the heat load was zero bar using the vacuum pump. After reaching steady operating state pressure inside the loop had a mean value of 0.24 bar. The rapid change of the pressure ranges between 0.02 and 0.12 bar. Volume flow rates were recorded for both crest and trough of the wave patterned pressure difference, where the undisturbed position was the wick capillary limit. The fact that pressure oscillation did not stop during this time period of 600 s indicates that the wick did not dry out because of vapor moving inwards.

Each time the pressure drop was smaller than the capillary limit of the evaporator, vapor flow was pumped toward the exit through the grooves, but each time the pressure drop was larger than the capillary limit of the evaporator, the vapor failed to flow through the structure of the porous wick. However, in the later case, the evaporator did not dry out because the pressure drop was reduced rapidly below the capillary limit and so vapor was now pumped to the condenser. At the same time, the temperature of the subcooled water, entering the evaporator and the porous wick, was adequate to cool the wick, allowing stable operation.

### 3D CFD MODEL OF THE EVAPORATOR

It has already been shown that a computational fluid dynamics (CFD) model could predict the heat transfer in the evaporator by Agerinos and Margaris [14]. Expanding that CFD model, a new three-dimensional (3D) CFD model was developed (Figure 6) in the Gambit software for geometry and mesh.
generation to analyze mass and heat transfer mechanisms in the porous wick inside the evaporator.

This computational model has a 3D-structured mesh of 3,650,000 cells. The mesh density was dictated in order to maintain the equilibrium between necessary computational time and accuracy. The purpose was neither optimum accuracy nor less time till convergence, but to design a model able to provide results of good agreement with the experimental ones in reasonable computational time. CFD Fluent, by ANSYS, was used, applying the mixture multiphase model and the SIMPLE algorithm for pressure–velocity coupling. In order to ensure numerical calculations convergence under-relaxation factors were utilized. Convergence was further examined by checking mass and energy balances.

For the solution convergence all residuals were set at $10^{-3}$, except for the energy that was set at $10^{-6}$. Regarding the under-relaxation factors values 0.2 were set for momentum and volume fraction and 0.5 for energy.

Boundary conditions utilized were velocity inlet, for flow inlet, pressure outlet, because mixture model was selected, hot wall, for external evaporator wall and porous media, for the evaporator wick of 0.7 porosity.

Evaporator wick permeability was $1.43 \times 10^{-11}$ m$^2$ and inlet diameter was 0.004 m. Similarly with the evaporator tested on the experimental set-up, 3D models were designed with two wick heated lengths, 20 and 40 mm, respectively. As described in the next section, the CPL experimental set-up used a wick made of copper with $80 \times 10^{-3}$ m length, $1.2 \times 10^{-4}$ m$^2$ transversal area and $3.17 \times 10^{-5}$ m pore radius. Temperature at the evaporator external wall was set at 373 K, equal to the temperature of the external wall measured at the experimental model.

4 COMPARISON OF 3D CFD PREDICTIONS WITH EXPERIMENTAL MEASUREMENTS

The pressure difference between the inlet and outlet of the evaporator for each volume flow rate recorded during the experiment was determined from the experimental data and compared with the capillary limit of the evaporator, as calculated from Equation (1). The evaporator CFD model could not simulate the continuous operation of the CLP evaporator, but could rather predict the above-mentioned pressure difference for the experimental values when the numerical solution converged for the given under-relaxation factors regarding pressure, energy and momentum for a constant volume flow rate each time. The key factor that determines whether the CPL can ceaselessly operate or not is the conservation of the equilibrium between the system pressure oscillation and wick capillary limit, as demonstrated in Figure 1.

Our research did not emphasize during the starting-up period, the necessary time for the working liquid to reach a steady temperature, measured at the evaporator outlet, when the heat load is applied at the evaporator external wall. After this period temperature at the evaporator outlet should remain constant, as long as the evaporator keeps pumping working liquid to the condenser. Experimental results, therefore, are not presented for this period. Our interest was mainly focused on finding a simple way to process all the experimental data recorded for the CPL steady-operating state; a snapshot is presented in Figure 5 pointing out that the pressure oscillations amplitude is not constant. Our DAQ system recorded values of the volume flow rate every 1 s for 24 h. After each experiment ended, for every value of the volume flow rate that has been recorded, a mean value of pressure drop between the evaporator inlet/outlet was calculated; figures of the evaporator inlet/outlet pressure vs. time are not presented. This way figures such as Figure 7 were created, illustrating the wick capillary limit during the normal operation and theoretical and experimental pressure oscillation. Subsequently, the pressure difference of the evaporator was calculated to be $\approx 200$ Pa, 30 Pa below the wick capillary limit. For the 20-mm wick heated length evaporator CFD model with water initial temperature of 20°C the divergence with the experimental pressure drop mean value was 0 Pa, for volume flow rates between 0.4 and 0.6 l/min, and 50 Pa for the rest of the values.

The predictions of the outflow temperature by the CFD model were found to be in good agreement with the experimental ones. When the CLP evaporator reached a steady-state flow condition, the measured temperature was 319 K, while the CFD
model predicted a flow temperature of 318 K, stable for vapor pipe tallness, i.e. a $<0.5\%$ difference.

Next case studied is Set-1b, where the agreement between computational and experimental results concerning the pressure difference was better for volume flow rates $>0.5\ l/min$, as shown in Figure 8. For volume flow rates $<0.5\ l/min$, the CFD model did not achieve a reasonably accurate prediction especially for very small volume flow rates, i.e. $\sim 0.1\ l/min$. The difference between the capillary limit and CFD model prediction was 60 Pa for very small volume flow rates and gradually reduced to 40 Pa for higher volume flow rates.

The CFD model performed better in predicting the temperature and heat flux inside the wick than previous efforts. The predictions were very close to the experimental measurements, as shown in Figure 9. The difference between the two curves is $<0.5\ K$, demonstrating the capability of the CFD model to predict the flow through the evaporator rather accurately, keeping in mind that the temperature sensor accuracy is 0.2 K.

A third case studied was Set-1c, with the CFD model predictions being in better agreement this time with the experimental measurements, as shown in Figure 10. The CFD model-predicted pressure drop was 50 Pa smaller than the capillary limit and very close to the mean value of the experimentally measured pressure drop. However, for volume flow rates $\sim 1\ l/min$, the CFD model predictions became less accurate, a behavior attributed to the rapid increase in the amplitude of the oscillations, which is much bigger than the amplitude of the pressure oscillations for smaller volume flow rates. Filling the loop with working fluid of initial temperature close to the saturation temperature caused quicker response of the system to the externally applied heat load. Fluid vaporization took less time and made the evaporator more sensitive to pressure oscillations. Consequently, more vapor could be produced inside the porous wick waiting to be pumped out of the evaporator when the pressure drop was smaller than the capillary limit. This behavior was also reproduced by the CFD model, as reflected by the shorter time needed for the numerical solution to converge.

In Figure 11, the experimental measurements of the outflow temperature are shown, together with the corresponding predictions of the CFD model. The difference between two curves was 1.4 K for the temperature near the upper wall of the evaporator, while this difference was decreased to 0.9 K near the bottom wall.

Next, the experiments with the 40-mm heated length of the wick are presented. The same cases were studied, as in the case of the 20-mm heated length, please see Table 1, in order to clearly demonstrate the influence of a longer wick heated length on the predictive capability of the CFD model.
The pressure oscillations at the evaporator inlet and outlet are shown in Figure 12. Comparing these results with the corresponding ones for the shorter wick heated length, i.e. Figure 6, it is observed that in the latter case the mean value of the pressure was 0.03 bar larger and the pressure decreased at a lower rate, starting at 0.3 bar and reaching gradually a value near 0.25 bar. Furthermore, the pressure at the evaporator outlet is clearly shown to be smaller than the inlet pressure, all the time, while this was not clearly shown in Figure 6. In summary, the overall operation of the CLP system was more predictable due to the fact that the porous wick was less sensitive to pressure changes, as a result of the larger heat flux incoming through the external wall of the evaporator.

The CFD model adequately reproduced the above-described behavior by accurately predicting the experimental measurements, as shown in Figure 13. It is important to mention that during the experiments it was observed that 50% of the heated length of the wick operated near the capillary limit. This behavior was faithfully reproduced by the CFD model which predicted a pressure drop very close to the capillary limit.

Furthermore, the capability of the CFD model to predict the temperature distribution at the evaporator exit remained the same, as shown in Figure 14, where the difference of the two curves is 1.5 K at the walls and 2.5 K along the centerline of the flow.

The predictive capability of the CFD model became even better for Set-2b, as shown in Figure 15. The CFD model was very reliable for volume flow rates <0.6 l/min, resulting in better prediction of the pressure oscillations.

The computed temperature curve was very close to the experimental one. Their difference was <1 K near evaporator wall and almost zero at the middle of the flow, as shown in Figure 16.

The final case was Set-2c, with the most extreme conditions tested in the present experiments, i.e. the fluid temperature was very close to saturation temperature and the heat transfer rate was higher than all the other tests. In this case, the predictions of the CFD model were better for volume flow rates >0.2 l/min, while it was slightly less accurate for volume flow rates <0.2 l/min, as shown in Figure 17.
Finally, the predictions of the CFD model of the temperature distribution were again very accurate, as in all other cases. Experimentally, an outflow temperature of 333 K was measured, while the CFD model predicted 332.9 K near the upper wall, 332.5 K near bottom wall and 332.3 K in the middle, as shown in Figure 18.

5 CONCLUSIONS

Experimental data concerning pressure oscillations of a capillary pump loop were found in good agreement with the predictions of a theoretical model describing the evaporator operation, for both types of evaporator tested.

Two computational 3D CFD models were designed and the flow through porous wick was computed. The predictions of both models were shown to be in good agreement with experimental measurements.

The CFD evaporator model with a 20-mm wick heated length and 293 K of water initial temperature predicted rather well the pressure oscillations for volume flow rates between 0.2 and 0.8 l/min. For 303 K predictions of the 3D model were very accurate for the whole range of experimental values of volume flow rate. Moreover, for 313 K the predictions were very satisfactory up to 0.9 l/min. In every case predictions were below the wick capillary limit. The computed outflow temperature presented a maximum difference of 1.5% compared with the experimental data, which is very satisfying. In more detail, for Set-1a the deviation between two curves was 1 K, for Set-1b from 0.2 up to 0.5 K and, finally, for Set-1c from 0.9 up to 1.4 K.

On the other hand, the predictions of the evaporator CFD model with a 40-mm wick heated length were even better, especially for a Set-2b case, because during the CPL operation the pressure distribution was smoother before vapor was pumped out of the wick each time. Additionally, the difference between temperature curves was from 0.8 up to 0.1 K. For Set-2a the prediction of the CFD model lost part of the accuracy only for a volume flow rate of 0.5 l/min. For Set-2c the same problem was faced only for 0.2 l/min. Eventually, predictions accuracy of the flow temperature at the evaporator exit for Set-2c was the same as Set-2b, while for Set-2a the temperature declination was slightly more, from 1.5 to 2.5 K.
ACKNOWLEDGEMENTS

This work has been co-financed by the European Union (European Social Fund (ESF)) and Greek national funds through the Operational Program ‘Education and Lifelong Learning’ of the National Strategic Reference Framework (NSRF)—Research Funding Program: Heracleitus II. Investing in knowledge society through the European Social Fund.

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