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# FULLY INTEGRATED ONE PHASE LIQUID COOLING SYSTEM FOR ORGANIC BOARDS

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## ABSTRACT

Prime concerns in designing liquid cooling solutions are performance, reliability and price. To that end a one-phase liquid cooling concept is proposed, where all pumps, valves and piping are fully integrated on board level. Only low-cost organic board technology and SMT processes are used in the design. This paper addresses the key issues of such a concept together with some numerical and first experimental results. It is highlighted that for such a concept a special type of membrane pump with adequate valve technology is especially suitable. Design guidelines as to its performance are given. Eventually, the obtained results are evaluated with respect to the requirements and necessary further developments are commented on to make the concept eligible for the cost-performance-sector.

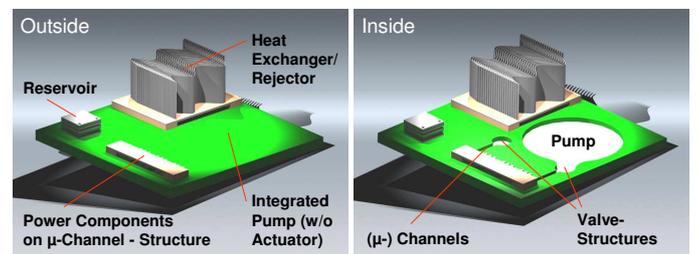
## 1. INTRODUCTION

As power and power-density of microelectronic components and devices have been continuously rising over the last years and are expected to keep doing so [1, 2], liquid cooling solutions remain an interesting but challenging alternative to air cooled solutions [3]. The advantages are obvious and well-known [4]. So are the disadvantages [5, 6]. Especially reliability and cost are key blockers against the large scale employment of liquid cooling solution for cost-performance, microprocessor or graphic-processing applications, whereas there is activity on the high performance sector [7, 8]. Perfect sealing being mandatory, it requires costly non-standard technology, sometimes a redundant pump circuit, lengthy reliability testing procedures and last but not least clearing of the psychological hurdle of combining water and electronics. Then, it requires a system approach: A liquid cooling system has to be customised to its application. Further, it requires many disciplines for reliable design, technology and test from a

physics, material science, engineering and packaging standpoint. This is what makes liquid cooling a challenge.

A one-phase closed-loop liquid cooling circuit consists of a cool plate for the power components, a heat exchanger to reject the heat into the environment and a pump and pipes to drive and contain the working fluid (most often water). Most presented concepts, however, feature a modular layout, requiring some kind of plumbing work during final assembly, giving rise to thermo-mechanical reliability concerns causing leakage [9, 10]. This is how the idea comes about to integrate all fluidic structures completely with conventional technology. Such a liquid cooling concept could look like figure 1. This concept is what this paper is about.

Where design and fabrication of micro-channels have been



**Fig. 1 Outline of the integrated liquid cooling concept. All piping and pump emerge during board manufacture and SMT process.**

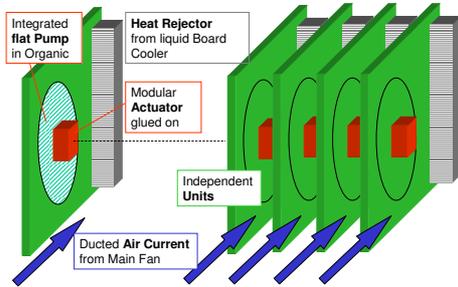
studied in detail, a central challenge for integration is the pump. Key requirements are: reliability, low cost, high performance (volume flow and pressure), low power consumption, small form factor, low noise. Pump designs have been proposed to meet these requirements, being modular though [11]. Pumps become interesting for electronics cooling exceeding a volume flow of  $\dot{V} > 20 \text{ ml/min}$  for a cooling power of  $P > 100 \text{ W}$  [12]. Pressures of  $p > 10 \text{ kPa}$  are desired but dependent on the overall fluidic resistance. These numbers serve for comparison to our concept which

is outlined below.

## 2. CONCEPT

The main idea is, that the pump does not actually exist as a modular entity. It evolves as the board is being manufactured together with all liquid-carrying channels and valve structures. So there are no open pipes and no necessary fluid connectors. The membrane is for the time being a thin copper-coated FR-4 sheet, sealing the pump. All these processes involve mostly standard (i.e. partially low-cost) or available tested processes during board manufacturing. Eventually a small actuator is simply placed and glued on top of the membrane and connected electrically. Micro-channels for the power-component bank or the heat-exchangers can be fabricated within the board also or soldered by solder-ring technology [13] onto the board (like eventually a liquid inlet or reservoir allowing filling after reflow). The complete containment of the liquid circulation within the board should guarantee reliable long term operation. The walls of all channels have to feature copper (or enhanced by some other metallisation during or after plating) to prevent water diffusion through the polymers.

The philosophy is to have a flat pump, not a bulky one, as often in a device there is unused space for a laterally extended component rather than a block, as would be the case for e.g. a time-honoured rotary pump. So pumps can have a diameter in the cm-range allowing higher volume flow at low frequencies. In this vein even a juxtaposition of many flat independent units is imaginable as depicted in figure 2, leaving enough space for the air to pass through.

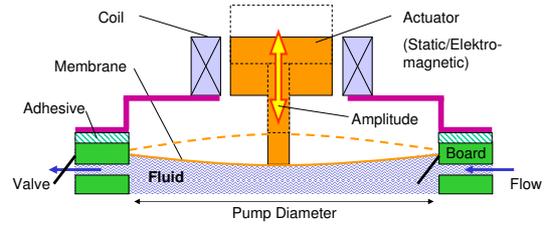


**Fig. 2 Possible arrangement of independent units for system cooling. As the units are flat, the air passes around them.**

Here, a ducted fan could remove the heat on system level. All fluid structures remain integrated, as each unit has its own liquid cooling circuit.

Using a membrane pump entails the use of some force to drive it. As a practical low-cost concept requires gluing of the actuator as a process step during assembly on board level (see figure 3), various actuator principles exist.

From piezo-actuators (high-voltage disadvantage, high price) or bucky-paper actuators (not yet mature technology), there remain electro-magnetic ones. (Other concepts

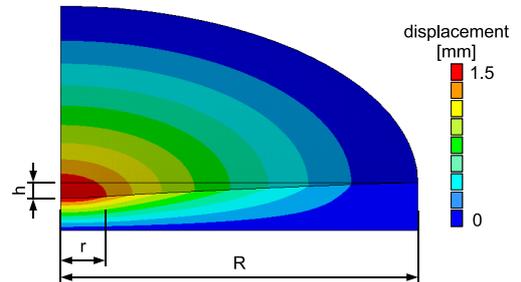


**Fig. 3 Schematic of pump with glued-on actuator. Further key features are valve and membrane characteristics.**

show not enough volume flow, see e.g. [11]). In the following we describe the design of the key elements of the membrane pump together with the valve-structures and evaluate them with respect to their performance as a first step to the proposed concept.

## 3. PRELIMINARY EXAMINATIONS

Usually membrane pumps consists of a pump chamber spanned with the membrane and appendant input and output valves. To estimate dimensions of the chamber for a given flow rate a simple truncated cone model was used as a first approximation of the displacement volume. Equation (1) can be used to calculate the required displacement  $h$  to get a given displacement volume  $V$ .



**Fig. 4 FE-model to estimate the displacement volume of an realistic membrane.**

$$V = \frac{h \cdot \pi}{3} (R^2 + R \cdot r + r^2) \quad (1)$$

By using FE simulations (see Fig. 4) a correction factor  $k$  could be found to improve the truncated cone model equation (2). Herby the deformation of a realistic membrane is expressed.

$$V_{disp} = \frac{h \cdot \pi}{3} (k \cdot R^2 + k \cdot R \cdot r + r^2) \quad (2)$$

$R$  is the chamber radius,  $r$  is the radius of actuator stamp and  $k = 0,65 \dots 0,75$ .  $k$  reduce the effective membrane area due to bending at rim.

An actuator is needed to move the membrane up and down. A simple FE-analysis shows the minimum force requirements to the actuator moving a membrane down.

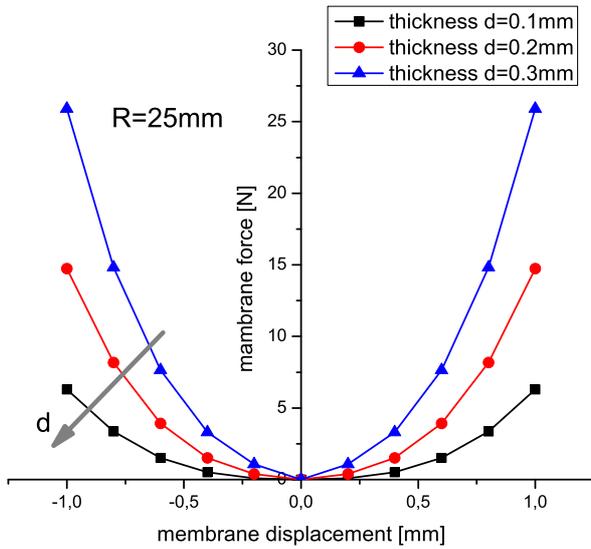


Fig. 5 Membrane force for different membrane thickness.

Several actuator concepts are examined. We found that only a stroke magnet with E-I core provides the needed force. To provide a force in both directions a spring was integrated (see Fig. 6).

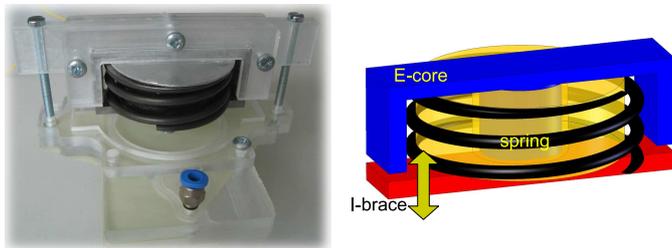


Fig. 6 Actuator with E-I core and spring to provide needed force in two directions.

#### 4. DESIGN AND EVALUATION OF PUMPING SYSTEM

As previously mentioned in and outlet valves are necessarily in membrane pumps. Several kinds of valves were examined and the best (V2 Fig. 8) one was chosen to design the prototypes.

##### CHARACTERISTICS OF VALVES

Valves are characterized by their characteristic curves where volume flow is plotted versus pressure drop. The

characteristic curve of a membrane valve is simply digital, assuming perfect diode functionality. Because of both forward (right) and backwards (left) flow in valves with non-moving parts the difference  $v_{eff} = v_{for} - v_{back}$  has to be evaluated (see Fig. 8).

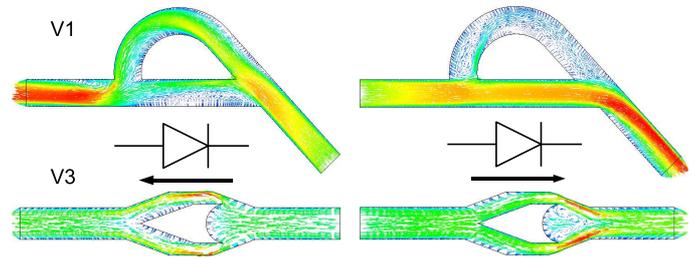


Fig. 7 Calculated velocity field in tesla valves. Flow resistance both for backward and forward flow can be derived.

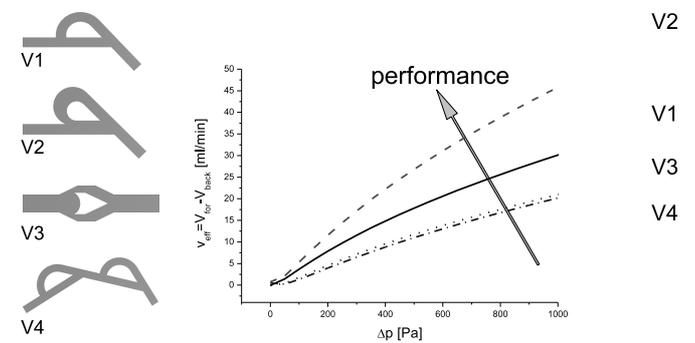


Fig. 8 Effective volume flow rate for different valve types (steady-state conditions).

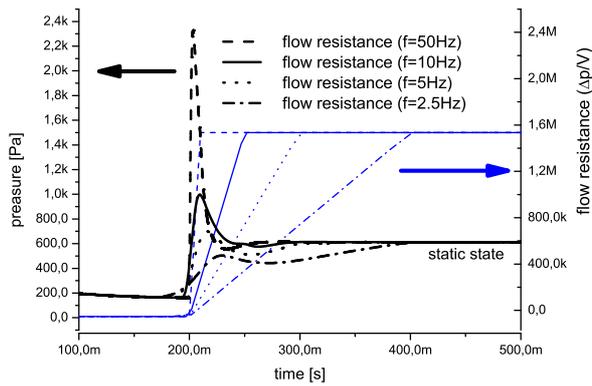
While pumping valves were alternately switched in closed and open mode. In case of switching inertia causes a significant increase in flow resistance. This effect was investigated using transient CFD analysis. Figure 9 shows the dynamic behavior of tesla valves. Different excitation frequencies of actuator up to  $f = 50$  Hz were simulated by using pressure steps with different slew rates. At higher frequencies ( $> 5$  Hz) an increasing valve performance is expected over the static case.

This is a fundamental result. It fully exploits the diode-effect of tesla valves, one needs to consider the dynamic behavior.

##### CHARACTERISTICS OF PUMP SYSTEM

Investigations of an entire pump system using CFD-analysis needs to consider different aspects. First the focus was on pump chamber and valves. Heat exchanger, compensating reservoir and i.e.  $\mu$ -channel cooler were not considered.

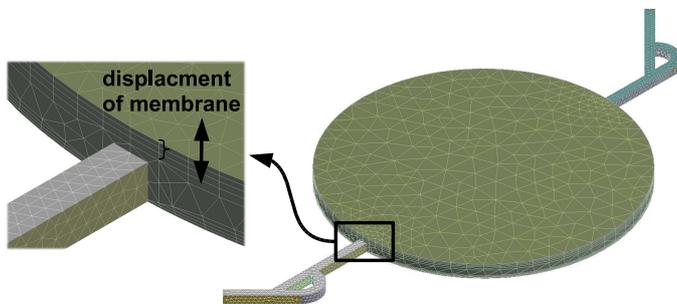
The moving membrane that drives the fluid is the main challenge in this model. One can do coupled field analysis



**Fig. 9 Dynamic behavior of tesla valves for different excitation frequencies (up to  $f = 50$  Hz).**

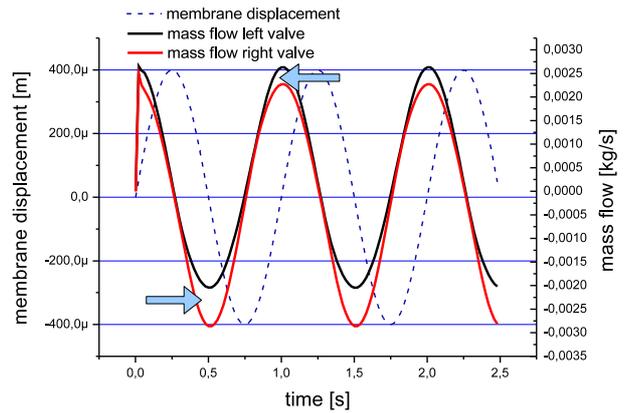
(fluid-structure) were physical domains interact with one other. This kind of analysis takes much time and computing performance. The moving mesh capability of ANSYS CFX10 can be used to generate a single domain (fluid) model of the membrane chamber. On the one hand one has to generate a mesh that can be compressed in one direction without critical deformation of the elements. This can be achieved by using prismatic element shape as shown in figure 10 on the left hand side. On the other hand you have to define a time dependent deformation (like Eq. (3)) of nodes located on membrane plane.

$$u = A \cdot \sin(2\pi \cdot f \cdot t) \quad (3)$$



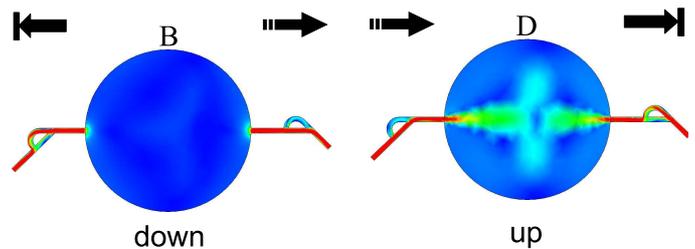
**Fig. 10 CFD analysis using moving mesh capability to simulate membrane displacement.**

Figure 11 shows the results of a CFD analysis. The membrane nodes was moved using (3) where  $A = 0.4$  mm and  $f = 1$  Hz as can see in dotted curve. The continuous curves represent the mass flow out of and into the valves. The different areas marked by arrows indicates an effective mass flow through the system. Integration over a period calculates the mass flow rate. Excitation with  $f = 3$  Hz using tesla valves V2 a performance of  $\dot{V} \approx 24$  ml/min was reached.



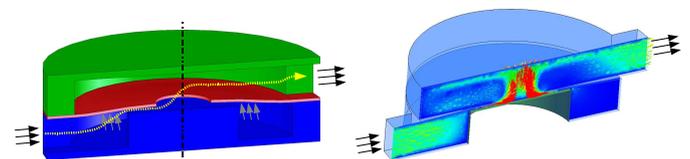
**Fig. 11 Simulation results: mass flow at inlet and outlet and sinusoidal membrane displacement with  $f=1$ Hz.**

Figure 12 shows different states of pumping cycle. To the left hand side fluid is pressed out while membrane moves down. On right hand side membrane moves upwards. The pump chamber fills up with fluid again.



**Fig. 12 CFD analysis of membrane pump with tesla valves.**

A second significant parameter to pump performance is the chamber diameter. A parameter study shows increasing flow rates for diameters up to 70 mm. In the range of 50-70 mm the flow rate behaves monotonic (cf Eq (2)). Extrapolation identifies an unrealistic diameter of approx. 200 mm to reach the desired flow rate of 100 ml/min. Although it is unrealistic to assume perfect membrane stiffness.



**Fig. 13 CAD model and CFD analysis of a membrane valve.**

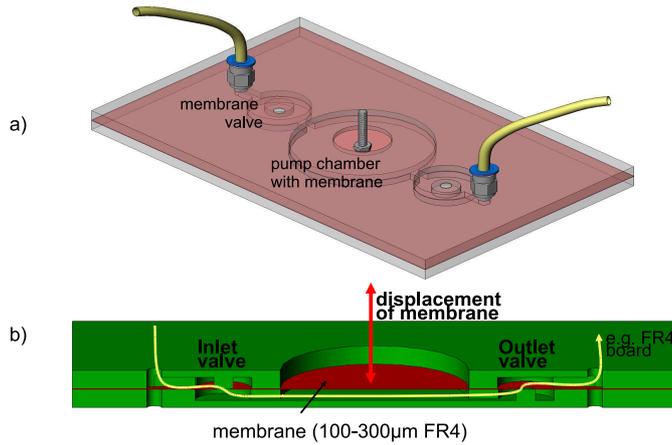
As a result, the pump design with tesla valves can not provide the desired performance. Although they are very

interesting assume reliability of the pump due to non-moving parts.

For this reason membrane valves were evaluated and used for prototypes and further investigations. Figure 13 shows an opened membrane valve on the left and flow behavior on the right.

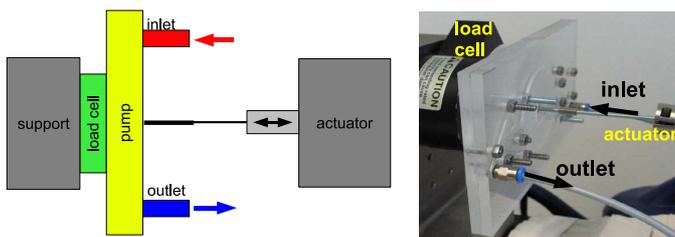
## 5. EXPERIMENTAL EXAMINATION OF PROTOTYPES

The first prototypes were not built in organic boards. The pump chamber and valves were milled in two PMMA panels as can see in figure 14.



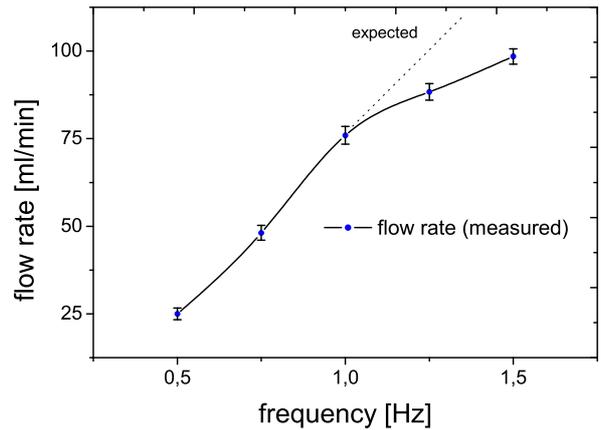
**Fig. 14** Prototype of pump a) with membrane valves, b) cross section of membrane pump.

For easy adjustment of excitation frequency and amplitude we used a dynamic testing system (*MTS Tytron 250*) as actuator. See principle setup and prototype under test in figure 15. So first evaluation as force measurement was possible.



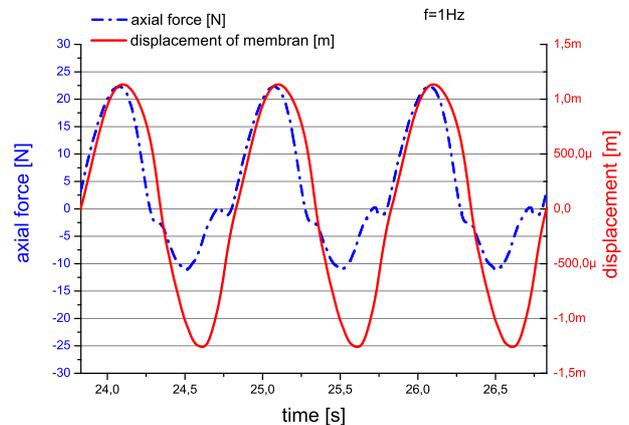
**Fig. 15** Test setup to measure flow rate of prototypes

Figure 16 shows the measured results of the pump with membrane valves and chamber diameter of 5 cm. Pump performance up to  $\dot{V} \approx 96 \text{ ml/min}$  was reached. The dimensions of the entire pump prototype are  $10 \times 7 \text{ cm}$  and  $\approx 10 \text{ mm}$  in height. The membrane was displaced  $A = \pm 1.1 \text{ mm}$  with  $f = 1 \text{ Hz}$ . See measured membrane displacement and force in figure 17.



**Fig. 16** Measured flow rate as function of exciting frequency (membrane valves).

During one pumping cycle a fixed quantity of fluid being displaced by the membrane. For that reason the flow rate should increase proportional to the excitation frequency. Difficulty in adjusting the right control parameter of microforce testing system causes a change of membrane displacement for different frequencies. It could be observed that membrane displacement decreases for higher excitation frequencies. Mass inertia of fluid and membrane restrict the flow rate.



**Fig. 17** Measured axial force and displacement while excitation with  $f=1 \text{ Hz}$

The control parameter for  $f = 1 \text{ Hz}$  could be found, as can see in figure 17. The membrane moves sinusoidally  $A = \pm 1.1 \text{ mm}$ . For negative forces (moving up) the curve shows irregularities caused by a spring-back of membrane. Measured membrane force is slightly higher than the simulated membrane force. By this, a small needed force to

displace the used membrane valves, is to be recognized. Nevertheless 20 N is too large and were not mainly used for pumping. The necessary actuator mentioned above (Fig. 6) seems not to be a economic solution because of no small form factor and heaviness.

## 6. CONCLUSION AND OUTLOOK

In this paper we have proposed a novel concept for the design of a fully integrated closed-loop liquid cooling system in organic boards. The individual key components and their principal role and make-up have been briefly discussed. As the heart of the concept the integrated pump has been discussed in detail using CFD and experiments.

It was found, that a membrane pump could be used to produce a volume flow which could be used for electronics cooling, depending on diameter, amplitude, frequency and above all, valve design.

We have shown that a membrane pump with chamber diameter of 5 cm and membrane valves is able to provide a pump performance of 100 ml/min. With simulations a effective flow rate of 24 ml/min could be shown of a pump with same chamber diameter and tesla valves. Therefore tesla valves seems not to be suited for these pump concept. Investigating the dynamic behavior shows the potential of tesla valves during higher frequency excitations. This is a fundamental result, one needs to consider the dynamic behavior.

Further steps toward system level include technological realisation of the liquid circuit in FR-4, using membrane valves, to assess pump performance on a real demonstrator. The membrane of the pump is to be thinned or substituted by a flat stiffened membrane with soft seam, as the present design uses too much energy for deformation. A drive similar to a loudspeaker coil may then be sufficient. Testing will first of all include hematicity and active thermal cycling.

It is believed, that such a design could contribute to enabling reliable and inexpensive thermal technology for liquid cooling concepts.

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