



Thermal study of a passive cooling device operating through a bubble lifting CLTPT of NOVEC 7000 with a two-fluid condenser

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ABSTRACT

In the framework of cooling technology applied to electronics, multiple setups and coolants have been used to achieve effective methods to remove high heat density fluxes. In this paper, an innovative device based on the combination of two fluids in a closed loop two-phase thermosiphon is proposed as an effective passive system to cool electronic components. The cooling device is an external flow boiling system that uses NOVEC 7000 as a circulating boiling fluid to transport heat from an evaporator capsule to a condenser. The bubble lifting effect in the closed-loop thermosiphon accelerates the fluid circulation and creates a natural jet impingement effect in the evaporator. In the condenser, the NOVEC bubbles ascend through a water pool, achieving a highly effective heat exchange between the fluids. The advantage of this two-fluid condenser system is highly efficient condensation since the vapour of NOVEC is condensed in a water pool, which produces a high heat transfer between the fluids and is not sensitive to the presence of non-condensable gases. In addition, the movement of the NOVEC bubbles produces high agitation in the water that enhances the heat transfer to the condenser walls. The system was tested under different working conditions with several power generation rates in the heat source and operating at atmospheric pressure and in a vacuum. The system was tested to assess the effect of water in the condenser, and heat transfer coefficients were calculated for the setup operating with and without water. In addition, a numerical model was developed to predict the different temperatures in the system and extrapolate the results for different operating conditions. Several tests were performed, and the results showed robust working of the system and good control of temperatures in the element to be cooled. The system was able to maintain a temperature lower than 70 °C in the aluminium block working with a heat flux of 220 kW/m² without any extra energy consumption. The effect of water in the condenser was shown to be effective in improving heat transfer and coolant condensation. The heat transfer coefficients in the condenser can be improved by 22 to 133%. The model for temperature prediction showed reasonably good results for most of the experiments regarding both transient behaviour and stabilization temperatures.

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1. Introduction

The continuous advances in the power electronics industry and the improvement of computer components have led to the need to develop more efficient cooling systems. High-performance elec-

tronic elements can be limited by the capacity to remove heat and lower the system operating temperature. Some studies report microprocessor cooling as the main limitation on the development of computation technology [1,2]. Therefore, there is a continuous challenge to increase the capacity and effectiveness for removing heat from electronic elements.

When classical air fans are insufficient to remove high rates of heat from electronic components, liquid cooling is a common solution to increase the cooling capacity [3]. There are several single-phase liquid cooling solutions, from simple liquid immersion to high thermal performance methods. Some of these methods were

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Nomenclature

A	area (m^2)
C_p	specific heat ($\text{J}\cdot\text{kg}^{-1}\cdot\text{K}^{-1}$)
d	diameter (m)
g	gravity acceleration ($\text{kg}\cdot\text{m}\cdot\text{s}^{-2}$)
h	heat transfer coefficient ($\text{W}\cdot\text{m}^{-2}\cdot\text{K}^{-1}$)
K	proportional coefficient (-)
m	mass (kg)
P	pressure (Pa)
$\dot{Q}$	heat transfer rate (W)
T	temperature (K)
t	time (s)
z	height (m)
<i>Greek symbols</i>	
α	linear equation coefficient ($\text{W}\cdot\text{K}^{-1}$)
β	linear equation independent coefficient (W)
λ	fraction of the evaporated NOVEC that reaches the water (-)
ρ	density ($\text{kg}\cdot\text{m}^{-3}$)
σ	surface tension ($\text{N}\cdot\text{m}^{-1}$)
<i>Subscripts</i>	
env	environment
t	total
th	Thermosiphon
$evap$	evaporation
l	liquid
v	vapour
in	incoming
out	outgoing
<i>Superscripts</i>	
0	initial value

reviewed by Agostini et al. [4]. A commonly used solution is single-phase cooling with microchannels, which is based on a high conductivity metal block, which operates as a heatsink, with several parallel channels of small hydraulic diameters (50–1000 μm). Several works [5–8] presented diverse geometries and materials focusing on parameters such as the heat removal capacity and the pressure drop through the heatsink. Prasher et al. [7] performed tests and were able to remove impressive values of heat flux (up to 1250 W/cm^2 in hotspots). Another choice to enhance the heat removal capacity of a cooling system is the use of a high conductivity porous media for the heatsink. Porous media provide a high heat exchange surface and create complex trajectories of the fluid streamlines. The porous structure commonly acts as a network of connected fins, which, if it is combined with a high conductivity material, the results in an efficient dissipative medium. There are two main types of porous heatsinks: sintering particles and metallic foams. Jiang et al. [9] compared the performance of porous particle media and microchannels in a heatsink and concluded that the microchannel heatsink has a better combination of thermal and hydraulic performance, as porous heatsinks offer better thermal behaviour. Zhang et al. [10] evaluated aluminium foam heatsinks and concluded that the optimum porosity was 0.8 and observed better thermal behaviour than that of the microchannel heatsinks for high pumping power values but worse thermal behaviour when the pumping power decreased. Bhattacharya and Mahajan [11] experimented on forced convection heat transfer in finned metal foam heat sinks with 0.9 porosity, various pore sizes and number of fins. The results show that heat transfer is significantly increased when fins are incorporated into metallic foam. In terms of pore size, for a given pressure drop, the heat transfer

coefficient (HTC) is higher in the larger pores (5 pores per inch), reaching an HTC of 1169 $\text{W}/\text{m}^2\cdot\text{K}$. Finned metal foam heat sinks outperform the longitudinal finned and normal metal foam heat sinks by a factor of 1.5 to 2, respectively. Another advanced cooling technique reviewed by Agostini et al. [4] is jet impingement cooling. Although this technique is commonly used with air cooling with metallic foam and metallic heatsinks [12], there are some studies that applied it for submerged water cooling [13,14]. This method presents a high ratio of power dissipated to power consumed, reaching values up to 1800 [4]. In contrast, the concentration of the heat dissipation within the impingement area can cause large temperature gradients within the cooled equipment. Therefore, complex constructive jet arrays are required to achieve more homogeneous heat dissipation.

In recent years, interest in electronic cooling has moved to the study of two-phase cooling. This refers to a particular liquid cooling in which the refrigerant evaporates when it is in contact with the hotter surface of the heatsink. This fact produces a great increase in the heat transferred due to the high velocity induced by the bubbles in the heatsink surroundings and the latent heat of evaporation. Numerous studies have been performed in fluid boiling for thermal management and heatsink design [15]. Similar to single-phase liquid cooling, there are several categories of two-phase cooling systems. The simplest one is pool boiling, in which the heatsink is submerged in a fluid that evaporates at a temperature in the operating range of the system. This was widely studied decades ago [16] and continues to be a field of interest due to the development of new fluids and geometries of the pool systems [17–20]. The main advantage of pool boiling is its passive operation since it does not need a pumping system. In contrast, pool boiling has lower cooling performance and reaches the critical heat flux (CHF) limit at lower heat fluxes than those of other configurations. A more advanced method is flow boiling, which applies a forced convective flux to heatsinks. This can present as direct or indirect configurations depending on whether the fluid is in contact with the component or not, respectively. Both methods are successful in the removal of high heat fluxes and avoiding a premature CHF. Several studies have deepened the study of flow boiling for a large variety of geometric configurations, fluids and surface treatments [21–23], although most of the studies have focused on the jet impingement effect for external flow boiling [24–28]. The jet impingement use for internal flow boiling is limited by the forces caused by the jet impact on the electronic components. In addition, during severe boiling, there is a risk of separation of the liquid layer from the impingement area [29].

A critical part of an evaporative cooling system is the condenser, since the heat transported by the fluid has to be efficiently removed to the environment to change the vapour to liquid and complete the closed cycle of the fluid. The condensers are usually divided into three main types depending on the external fluid type. Hosoz and Kilicarslan [30] compared the performance of air-cooled, water-cooled and evaporative condensers and found a better behaviour of the water-cooled system. The efficiency of condensation also depends on the heat transfer performance on the sides of the condensing fluid, which is commonly enhanced through the surface treatment of the heat exchangers [31]. An important parameter in the heat transfer performance is the presence of non-condensable gases, which drastically reduces the HTC of the condenser surface [32,33]. This effect is highly pronounced in passive systems, since non-condensable gases make the natural circulation that is crucial in these systems difficult [34].

An important field to consider in the evolution of cooling technology is the field of working fluids. There is a wide range of fluids commonly used in cooling from traditional coolants to dielectric fluids. Water clearly presents superior thermal properties in comparison to any other fluid for single phase indirect cool-

ing. The high boiling temperature of water makes it inappropriate for two-phase cooling in electronic devices at atmospheric pressure, and its thermal conductivity limits the use of systems in which the fluid is separated from the electronic component. This can be attached to indirect cooling systems operating at a near-vacuum pressure [35,36]. These problems have made dielectric fluids a popular option as working fluids for two-phase cooling. Several works were carried out in pool boiling cooling with Freon-type chlorofluorocarbon (CFC) fluids [37–39]. These fluids were considered ideal fluids for two-phase cooling due to their stability properties and low boiling point, but their use has been gradually replaced since they were proven to be highly harmful to the ozone layer. In recent years, interest has shifted towards other refrigerants, such as fluorocarbon (HFC) and hydrofluoroether (HFE) fluids, which present the same desirable properties as Freon-type fluids and offer considerably better environmental behaviour. These fluids present a high wetting behaviour due to their low viscosity, which provides good contact and convective effects for any kind of surface [40]. Multiple works have evaluated the heat transfer of HFCs [41–43] and HFEs such as NOVECs [44–46]. In particular, NOVEC 7000 is of particular interest due to its low boiling point (34 °C at atmospheric pressure), which is quite appropriate for the operating range of electronic devices. Another type of fluid broadly studied is nanofluids, which are fluids with nanoparticles in suspension uniformly mixed with the base, providing an increase in the heat transfer capacity of the fluid and, particularly, of its CHF [47–50]. Kristiawan et al. [51] found that TiO_2 particles enhance convective heat transfer for both laminar and turbulent flows. They also combined this technology with enhanced surfaces and found that nanoparticles produce an increase in heat transfer on microfin tubes [52]. The enhancement effect of nanofluids in the boiling regime was studied by Kim et al. [53], who found a significant enhancement for different fluids with low concentrations of silica, alumina and zirconia nanoparticles. Kwark et al. [54] investigated the nanoparticle deposition mechanism and found an increase in the surface wettability and CHF.

Another field of study for the possible improvement of the cooling behaviour of heatsinks is related to the surface parameters. The most studied parameters are the surface roughness and the contact angle. Jones et al. [55] found an important increase in heat transfer when the surface roughness was increased for FC-77; however, this effect was not observed for water. Chang and Lin [56] observed the suppression of boiling instabilities, improvement in boiling heat transfer and reduction of the cooling power required when the evaporator surface is scaled-roughened instead of smooth-walled. El Genk and Pourghasemi [46] also found a notable increase in nucleating boiling heat transfer and CHF for NOVEC 7000 when increasing the surface roughness. Other surface effects that have an important influence on pool boiling heat transfer are the surface orientation, ageing and surface structures, as Fan and Duan [15] reviewed.

This paper focuses on passive two-phase immersion systems. This technology showed great cooling potential to remove a high heat density in the server electronics industry [36]. The most remarkable characteristic of these systems is the combination of high heat removal capacity with simple construction and lack of pumping, which makes them a highly energy efficient option. Although the two-phase cooling systems are characterized by a high heat transfer capacity at the elements to be cooled, equally important is the ability to transport and evacuate the heat out of the system. In this sense, closed loop two-phase thermosiphon (CLTPT) devices have been proven to be an effective solution [43,57,58]. They work with the principle of buoyancy flow to thermally drive the hot fluid from the heat source (evaporator) to a heatsink in a closed circuit. Multiple types of CLTPT experimental devices can be found in the literature, as Franco and Filippeschi [58] reviewed. Neverthe-

less, they propose a general classification of three major types of thermosiphons: single-phase thermosiphons, CLTPTs with bubble lift and CLTPT-gravitational channel heat pipes. The first circuit is a simple circuit of liquid circulation due to natural convective flow. In the second type, the closed loop is full of circulating liquid, and liquid transport is accelerated by the bubbles that travel from the evaporator to the condenser. This effect produces relatively high fluid velocities, considering that no fluid pumping is induced. The third type is also a CLTPT, but only a part of the loop contains liquid. The rest is occupied by the vapour that travels to the condenser. In this type of CLTPT, it is critical to determine the optimal fill charge since an insufficient amount of working fluid in the evaporator liquid chamber causes deterioration in the heat transfer and system thermal performance. However, increasing the working fluid fill charge increases the liquid film thermal resistance and consequently reduces the evaporation heat transfer [59]. Several studies have focused on the relationships between the mass flow rate and geometrical parameters such as the raiser pipe diameter [60–63]. Khodabandeh and Furberg [43] studied the influence of different channel geometries on the heat transfer and flow regime with the R134a fluid. They found that HTC is enhanced and flow instability decreases when the height of the channel in the evaporator is increased. As said above, the presence of non-condensable gases is an efficiency reduction factor in condensers that has an important effect on passive systems. This affects the CLTPT, which was investigated by Burgazzi [34], who pointed out the gas load and radial gas diffusion as the main significant parameters for this heat transfer reduction.

This paper presents a novel CLTPT cooling system aimed at improving the cooling effectiveness of electronic components with a reduced energy cost. This innovative system is based on a loop thermosiphon, with an evaporator operating in a flow boiling regime, and the combination of three fluid phases (vapour and liquid NOVEC and water) in the condenser. The aim of this system is to improve the heat transfer efficiency in the condenser and avoid the negative effect of non-condensable gases. The thermal behaviour of the system is evaluated by performing different tests where the heat source to be cooled is an aluminium block. The evolution of temperatures in the principal components of the device is monitored and analysed under a range of heat powers applied in the heat source and different pressure conditions. In addition, a numerical model is proposed to predict all the measured temperatures in the system and to extrapolate the results to conditions other than the experimental conditions.

2. Experimental system

2.1. System design

Several parameters need to be taken into account in the design of a CLTPT. From the system geometrical dimensions to the material physical and thermal properties, the most influential properties are those affecting heat transfer and fluid circulation forces. Among them, a crucial parameter that affects the flow circulation and the boiling regime is the diameter of the raiser pipe. This diameter can determine what type of force, capillarity, gravity or inertia is the predominant in the circulation of the fluid along the loop. These forces can be compared through dimensionless numbers, such as the Weber (inertia/capillary), Froude (inertia/gravity) and Bond (gravity/capillary) numbers, as summarized by Franco and Filippeschi [58]. A Bond number (Bo) of the order of 4 determines the threshold to allow bubble development in the pipe or a plug/slug effect. From this result, a minimum critical diameter is derived to allow bubble buoyancy lift. In a similar way, a maximum critical diameter determines whether the bubbles are confined in the pipe or not. This fact implies that bubble lift produces

Table 1
NOVEC 7000 Thermal and physical properties [64].

Parameter	NOVEC 7000 (C ₃ F ₇ OCH ₃)
Boiling Temperature (°C)	$\frac{3548.6}{22.978 - \ln[P/(Pa)]} - 273.15$, (34 °C at 1 atm)
Latent heat of vaporization (kJ/kg)	142 (at 34 °C)
Thermal conductivity (W/m·K)	$0.0798 - 0.000196 \cdot T(^{\circ}\text{C})$
Liquid density (kg/m³)	$1472.6 - 2.88 \cdot T(^{\circ}\text{C})$
Specific heat (J/kg·K)	$1091.9 + 0.3419 \cdot T(^{\circ}\text{C}) + 0.0039T^2$
Liquid viscosity (cSt)	0.32 (at 34 °C)
Surface tension (mN/m)	10.8 (at 34 °C)
Coefficient of expansion (K⁻¹)	0.00219
Molecular weight (g/mol)	200

a pumping action if they are confined in the riser pipe and that a flow boiling regime takes place. In contrast, for higher pipe diameters, the bubbles rise separately, and no two-phase lift is observed. Therefore, the evaporator operates in a pool boiling regime. An approximated maximum critical diameter was estimated by Chrisholm [63]. Both minimum and maximum critical diameter estimations are shown in Eq. (1). This diameter is dependent on the fluid thermal properties, which are summarized in Table 1.

$$d_{crit_min} = 2 \cdot \sqrt{\frac{\sigma}{g(\rho_L - \rho_V)}}, \quad d_{crit_max} = 19 \cdot \sqrt{\frac{\sigma}{g(\rho_L - \rho_V)}} \quad (1)$$

Taking these results into account, an experimental setup was prepared to achieve efficient cooling through a confined flow boiling evaporator, a closed-loop thermosiphon and a novel condenser that combines NOVEC and water. This device is able to transport heat across three fluid phases from a hot spot to the condenser heatsink without any pumping device, which means no energy consumption. The hot spot is made of an aluminium block with an electric resistance inside. The aluminium block is in contact with a capsule containing NOVEC, which operates as an evaporator. This capsule is connected, through a couple of hoses, to a large vessel filled with a mixture of NOVEC and water. The vessel is an aluminium cylinder with a finned heatsink that works as a condenser. The total external area of the heatsink is 1.27 m². A scheme of this setup is shown in Fig. 1.

As the NOVEC density is higher than that of water, NOVEC is located at the bottom filling the evaporator capsule in contact with the heat source. In this way, when the temperature of the hot spot is higher than that of the NOVEC evaporation point, the vapour generated ascends through one of the hoses to the vessel, and liquid NOVECs descends through the other hose, creating a closed-loop circulation between the capsule and the vessel. To achieve this, a slight inclination of the capsule is enough to place one of the hoses above the other. It is also necessary that the water-NOVEC mixture in the condenser vessel is located over the NOVEC evaporator capsule. For NOVEC 7000, values of approximately 1.9 and 18 mm are calculated for the minimum and maximum critical diameters, respectively. The internal diameter of the hoses that connect the evaporator and the condenser vessel is 12 mm, which ensures a flow boiling regime in the evaporator and thus a natural pumping action. In the water-NOVEC mixture, vapour NOVEC is lifted through the water zone and releases heat to water. This increases the water temperature and releases heat to the environment through the heatsink until an equilibrium temperature is reached. This system allows operation with a low quantity of NOVEC and uses the higher heat capacity and thermal conductivity of water in the condenser heatsink. The whole system is completely passive since it works through the buoyancy flow of vapour, which means there is no need for any pumping or external action.

Several thermocouples were located in the system to measure the temperature and pressure at different points. After several tests, it was proven that a high vapour stream causes great fluid movement and turbulence, which homogenizes the temperatures

of each fluid. Therefore, a thermocouple is necessary in each fluid to register the temperature evolution. The first thermocouple is located inside the aluminium block (heat source). This temperature is important since this element represents the element to cool. The second thermocouple is located at the bottom of the vessel (2 cm over the bottom) to monitor the temperature of the NOVEC. The third thermocouple is located 15 cm under the top of the vessel to measure the temperature of the water. These are the main temperatures that were monitored and are shown below to analyse the thermal evolution of the system during the experiments. Other thermocouples were used to measure the temperatures of the walls and the interior of the condenser as shown in Fig. 9. Two manometers were located in the evaporator and on the top of the condenser but only to ensure that the working pressures were in agreement with the evaporation and condensation temperatures. In addition, a valve is located on the condenser top to evacuate the system and perform some experiments at low pressure.

The heat source is located in an aluminium tablet 8 cm in diameter and 1.5 cm high. The vessel that works as the NOVEC condenser has an inner diameter of 10 cm and is 50 cm high. Its thickness is 5 mm, and it is surrounded by vertical fins along its entire height. The entire system is charged with 0.5 kg of NOVEC 7000 and 3 kg of water.

The operation conditions chosen for the experiments are the environmental conditions under which an electronic device commonly works. A temperature of approximately 20 °C was used as the environmental and initial temperature of the elements and fluids for all the tests. Two sets of tests were performed at atmospheric pressure and under vacuum at the initialization of the experiments. In the tests, the initial pressure is 0.1 bar over the saturation pressure of NOVEC 7000 at the initial temperature, which means approximately – 0.4 bar. The power inputs used in the heat source rank between 55 and 300 W (40.3 to 220 kW/m²). These are values of heat densities commonly found in computer processors.

The vessel is tightly closed to avoid NOVEC losses. As the NOVEC evaporation temperature changes with pressure, an air stopcock is located at the top of the vessel to evacuate the air and test the system working with low pressure. The set of electrical resistances, the aluminium block and the NOVEC capsule were insulated during the experiments performed to avoid heat losses in that area.

2.2. Effect of the two-fluid condenser

As the densities of water and NOVEC are remarkably different, the solubility of NOVEC in water is virtually zero [64,65]. The heat exchange between both fluids can be produced by conduction through the fluid interface, but when there is NOVEC evaporation, a greater fraction of the heat exchange is produced by bubble movement across the fluids. As the NOVEC viscosity is very low [64], fine bubbles are produced inside the liquid NOVEC phase. These bubbles ascend to the fluid interface, and when an sufficient

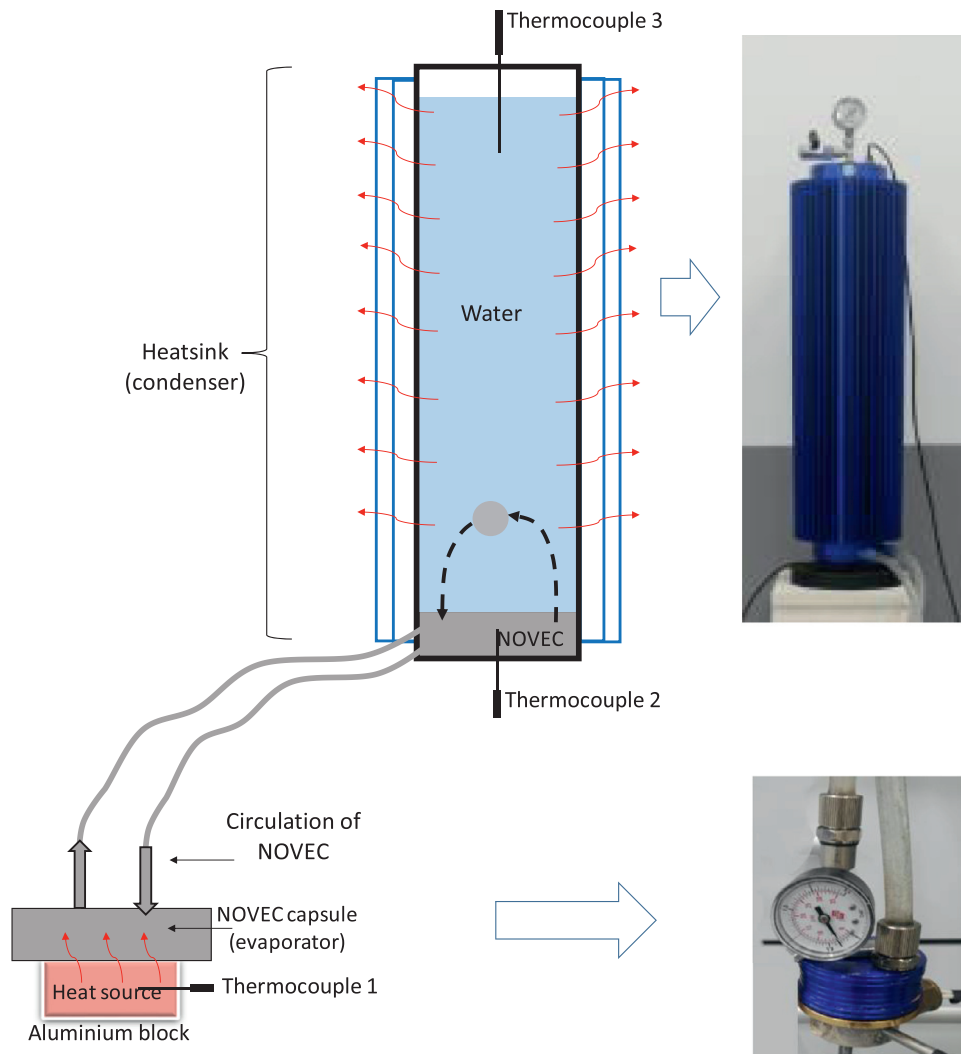


Fig. 1. Schematic of the experimental setup.

number of bubbles has accumulated at the interface, a large bubble (2–3 cm of diameter) of NOVEC vapour penetrates the water interface and ascends through the water. Then, almost instantaneously, this NOVEC bubble is condensed, and a liquid NOVEC drop is generated and falls on the liquid NOVEC phase located at the bottom of the vessel. This process occurs in a short time (approximately 1 second) and within a few centimetres of space. Therefore, the heat exchange between the NOVEC bubbles and the water can be considered virtually instantaneous. In addition, the movement of the NOVEC bubbles produces a great stir inside the water, which homogenizes the temperature of the whole water volume. Fig. 2 shows a capture of both the evaporation and the condensation process of NOVECs in a transparent vessel to visualize the process. In this case, the water is dyed green to distinguish it from NOVEC. As reported above, high agitation of both fluids, water and NOVEC was observed.

A novelty of the proposed looped thermosiphon system is the direct condensation of vapour NOVEC 7000 in a water pool. The condensation of vapour in a liquid pool is a phenomenon broadly studied by numerous authors and for different systems [66–72]. This produces a high heat transfer between fluids, the heat transfer coefficient is in the range between 10^5 and 10^6 (W/m²·K) for water [66,67], and a large exchange surface. This means that the heat transfer between NOVEC and water is very efficient. The heat transfer between the water and the condenser walls is also effi-

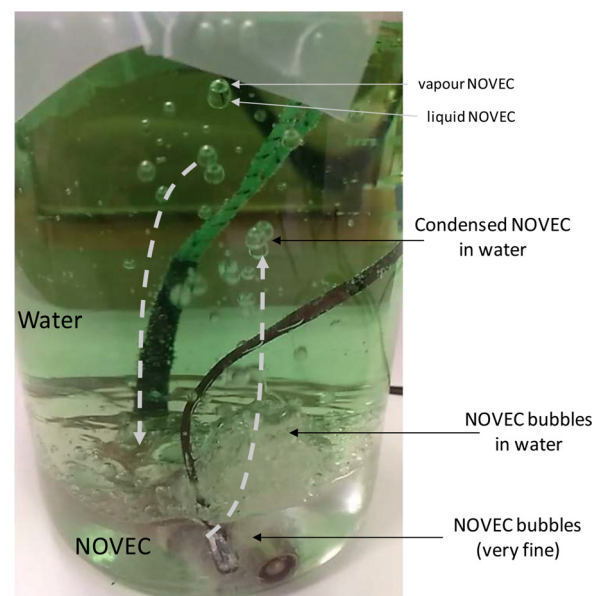


Fig. 2. Behaviour of the NOVEC and water mixture during NOVEC evaporation.

Table 2
Accuracy of measurement instruments and uncertainty of parameters.

Instrument	Accuracy of measurements
Thermocouples	± 0.1 (K)
Power input	± 0.1 (W)
Power input oscillation	< 2 (W)
Vacuum gauge	± 0.025 (bar)
Pressure gauge	± 0.1 (bar)
Parameter	Maximum uncertainty
$h_{\text{condenser}}$ (W/m ² ·K)	13.7%
R_{th} (K/W)	47.1%
R_{overall} (K/W)	3.7%

cient since the water is highly stirred. In addition, it is well known that the presence of non-condensable gases drastically reduces condensation heat transfer. Less than 1% of air in a water steam condenser reduces the HTC by half [73]. Nevertheless, in the two-fluid condenser, the effect of non-condensable gases barely has any effect since most of the heat transfer takes place through the water to the condenser walls. The HTCs inside the condenser working with and without water were also calculated and are shown below to analyse the effect of the improvement in the system heat transfer.

2.3. Uncertainty analysis

The accuracy of the measurement instruments is shown in Table 2, as well as the maximum uncertainty of the HTC and the thermal resistances of the setup. The thermocouples used to register the temperature evolution of the system elements have an accuracy of 0.1 K, which is precise enough to analyse the thermal behaviour of the system. However, this accuracy affects the uncertainty of other parameters. The most noticeable lack of accuracy comes from the power input since its value is not stable and it requires constant vigilance to avoid a high deviation from the set point. Thus, even though some oscillations could happen during the tests, they were monitored to ensure that there was no oscillation greater than 2 W. The pressure gauge measurements were not recorded during the experiments. They were only used to check the vacuum at the beginning of the experiments and to ensure that the pressure of the system matches the NOVEC saturation pressure.

The accuracy of the measurements also affects the HTC and the thermal resistance, as shown in Section 3. Eqs. (2–4) show the relative value of the maximum uncertainty of the HTC and thermal resistance of the system, respectively. The maximum values reached by these are 13.7% and 47.1%, respectively.

$$\frac{\delta h}{h} < \left[\left(\frac{\delta \dot{Q}}{\dot{Q}} \right)^2 + \left(\frac{\delta T_f}{T_f - T_w} \right)^2 + \left(\frac{\delta T_w}{T_f - T_w} \right)^2 + \left(\frac{\delta A}{A} \right)^2 \right]^{1/2} \quad (2)$$

$$\frac{\delta R_{\text{th}}}{R_{\text{th}}} < \left[\left(\frac{\delta \dot{Q}}{\dot{Q}} \right)^2 + \left(\frac{\delta T_{\text{water}}}{T_{\text{evap}} - T_{\text{water}}} \right)^2 + \left(\frac{\delta T_{\text{evap}}}{T_{\text{evap}} - T_{\text{water}}} \right)^2 \right]^{1/2} \quad (3)$$

$$\frac{\delta R_{\text{overall}}}{R_{\text{overall}}} < \left[\left(\frac{\delta \dot{Q}}{\dot{Q}} \right)^2 + \left(\frac{\delta T_{\text{env}}}{T_{\text{block}} - T_{\text{env}}} \right)^2 + \left(\frac{\delta T_{\text{block}}}{T_{\text{block}} - T_{\text{env}}} \right)^2 \right]^{1/2} \quad (4)$$

The system studied in this work has a high thermal inertia, which produces a smooth evolution of the measured parameters.

In this way, any instantaneous change or oscillation in the operating conditions or the surroundings is damped by the system. As said above, the most important instability that can affect the measurement accuracy is the oscillation of the power input. In any case, the maximum oscillation value found in the thermocouples when the steady state was reached was 0.1 K, which is the accuracy of the thermocouples used. This fact shows the effect of the system thermal inertia.

3. Experimental results

In this section, the results of several experiments performed in the system are shown. First, a mixture of water and NOVEC 7000 is tested at atmospheric pressure at the beginning of the experiments. The system is tested by applying different power values in the heat source. Then, the same experiments were performed after evacuating the condenser vessel to produce a vacuum, which decreased the evaporation temperature of the NOVEC. Finally, some experiments are performed with water liquid cooling (without NOVEC) to compare against the performance of the NOVEC-water mixture cooling system.

3.1. Tests at atmospheric pressure

Several experiments were performed with a low amount of NOVEC 7000 and pure water in the condenser vessel by changing the power generated in the heat source. The power varies in from 55 to 300 W. The experiments start from an initial environment temperature of 17–18 °C in the whole system, and the temperatures of the heat source, NOVEC fluid, water, and environment are registered up to equilibrium. Fig. 3 shows the results of the experiments with 100, 200 and 300 W generated in the heat source.

In general, the temperature evolution follows the same patterns observed in other reference works on passive liquid cooling systems [74,75]. All experiments show an aggressive increase in the temperatures of the heat source and the NOVEC liquid at the beginning and then soft heating of both. Water heating is slower since its thermal inertia is clearly higher and the mass of water is predominant. A delay of several minutes is observed in the water heating starting that matches the time that NOVEC needs to reach the evaporation temperature (34 °C). This fact shows that vapour movements are the predominant heat transfer mechanism between both fluids. The most important value to analyse the cooling efficiency of the system is the equilibrium temperature of the heat source. The experiment with 300 W of heat power is stabilized when the heat source reaches a temperature of 82.5 °C, which is a reasonably low temperature since the heat density in the contact of the aluminium block and the NOVEC capsule is approximately 220 kW/m². An interesting fact is that the temperature difference between NOVEC and water is lower as the power of the heat source increases. This result is because when power increases, a higher fraction of the heat transferred to the NOVEC is applied to evaporate the NOVEC, and the bubbles reach the water and transfer the heat directly to the water instead of heating the liquid NOVEC.

Despite the high thermal inertia that the water gives to the system, the heating of the aluminium block is fast. In approximately five minutes the temperature is approximately 90% of the final stabilization temperature. From that point on, the temperature difference between the aluminium block and NOVEC is virtually constant. This is caused by the thermal resistance between both elements. Something similar occurs with NOVEC and water, whose temperature difference is nearly constant during the whole heating process. Finally, the final temperature difference is imposed by the thermal resistance from the heatsink of the vessel to the air and

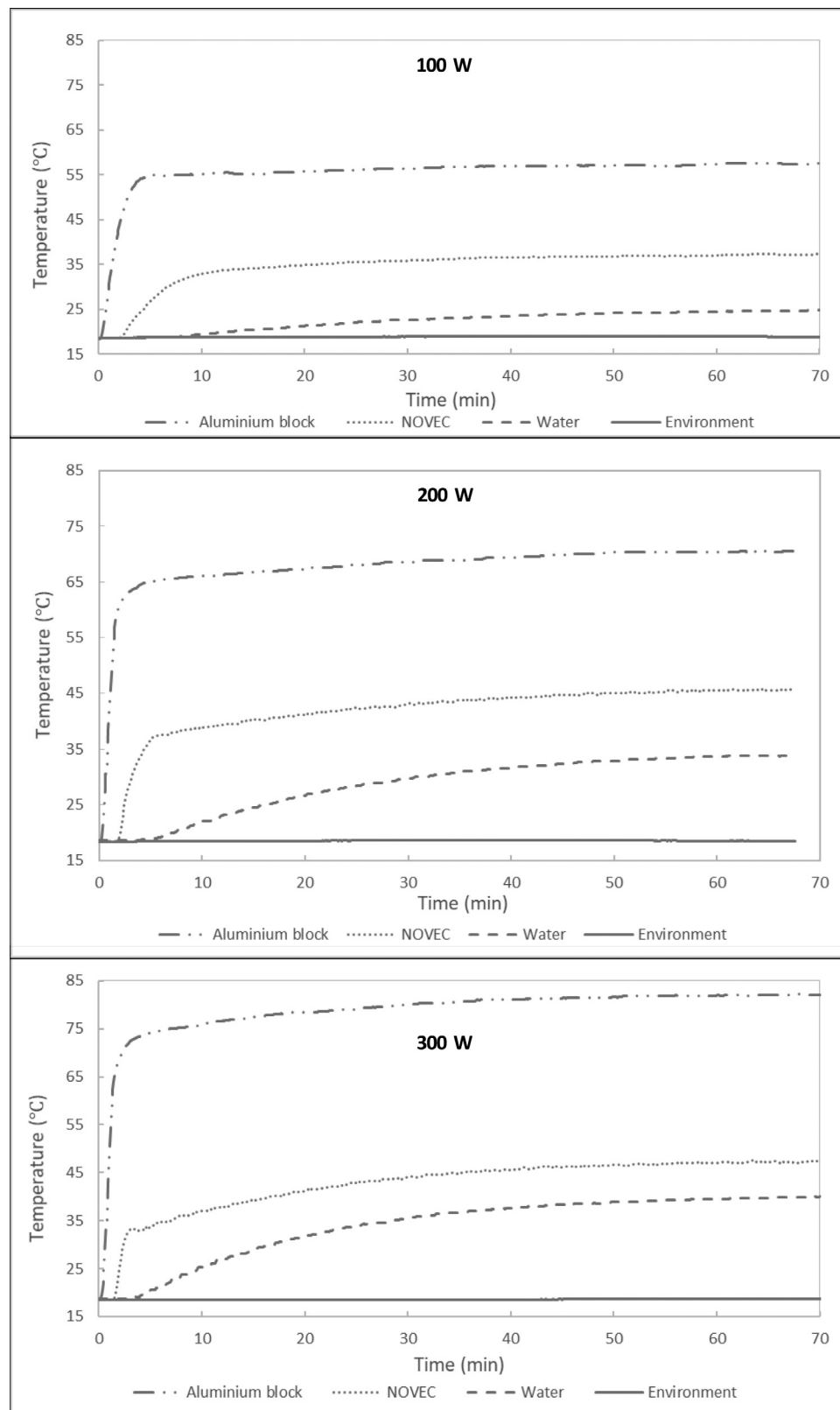


Fig. 3. Temperatures of the heat source, NOVEC fluid, water, and environment for experiments with 100, 200 and 300 W in the heat source and atmospheric pressure initialization.

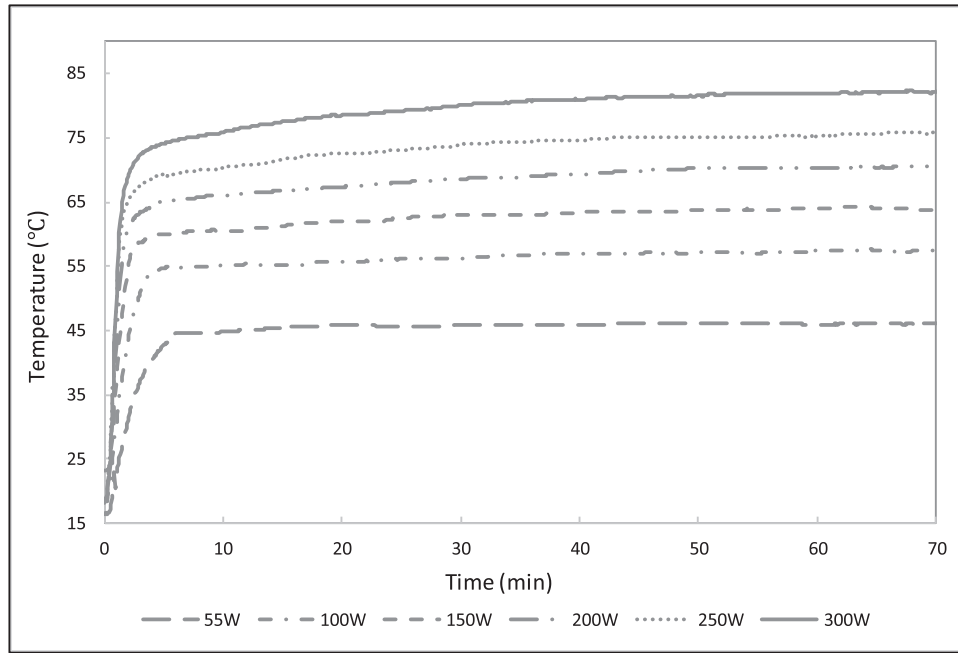


Fig. 4. Temperatures of the heat source for different heat source values from 55 W to 300 W and atmospheric pressure initialization.

the external air temperature. Therefore, the thermal resistances between elements and fluids and the water thermal inertia mark the evolution and the final state of the system.

The temperature of the heat source (aluminium block) is the most important temperature since the goal of the system is cooling the aluminium block that models a hot electronic device. Fig. 4 shows the temperatures of the aluminium block for different heat source values in a range of 55–300 W. As the diameter of the aluminium block is 41.5 mm, the heat transfer density ranks between 40.6 and 221.8 kW/m². These results show quick heating and stabilization of temperature and reasonably low final temperatures for all the measured experiments. It is important to mention that, as the system is completely passive, these results can be considered positive since the system operates without any energy expenditure. Compared with a similar loop cooling system with a fluid reservoir [76,77], the range of temperatures of the heat source can be considered low, which can be related to efficient overall cooling.

Fig. 5 shows the thermal resistances of the thermosiphon and the overall systems. On the one hand, the thermosiphon resistance considers the fluid temperature difference, from the NOVEC bubbles, calculated as the evaporation temperature, to the water thermocouple located in the upper part of the condenser. On the other hand, the overall thermal resistance considers the maximum temperature difference, which means the heat source (block) and the environment. Both are shown in Eq. (5). While the overall thermal resistance shows values in a range of 0.2–0.5, which are typical values of closed loop thermosiphon systems [76,77], the thermosiphon (fluids) thermal resistance shows lower values in the range of 0.02–0.1, which relates to the efficiency of the fluid circulation. The thermal resistance shown in Fig. 5 follows the simple formulation of Eq. (5), which is the ratio of temperature differences and the power input. Both thermal resistances decrease with the power input due to the higher vapour mass flow rate produced by the heat flux that stirs the fluids and improves the heat transfer between fluids and the HTC of the evaporator and the condenser walls.

$$R_{th} = \frac{T_{evap} - T_{water}}{\dot{Q}}, \quad R_{overall} = \frac{T_{block} - T_{env}}{\dot{Q}} \quad (5)$$

3.2. Tests at low pressure

The evaporation temperature of all liquids is dependent on pressure. The NOVEC 7000 evaporation temperature decreases when the pressure drops (Eq. (6) [64]). This fact allows an increase in the cooling capacity of the system by decreasing the internal pressure. To test this effect, the same experiments that were performed at atmospheric pressure were later performed by removing the air in the vessel at the beginning of the experiments. The vacuum pressure of these tests is 0.1 bar over the NOVEC saturation pressure at the initial temperature (approximately 0.5 bar). When the vacuum pump decreases the total pressure below this value, the NOVEC boils until its vapour pressure is equal to the saturation pressure. Therefore, lowering the pressure beyond this value has no effect. When the vacuum gauge slightly exceeds the saturation pressure, the evacuation is over, and the experiment is ready to begin. Then, as the experiments progress, the internal pressure depends on the vapour pressure equilibrium between NOVEC and water. The pressure in the evaporation zone of the system is the sum of the gas partial pressures and the column of both liquids, as shown in Eq. (7). Therefore, if P_{air} is considered vanishing, the values are easily calculated if the gas temperature is known. The air temperature can be assumed to be the temperature registered by the thermocouple located in the water top region.

$$T_{evap} = \frac{3548.6}{22.978 - L(P_T)} \quad (6)$$

$$P_T = P_{air} + P_{H_2O} + P_{NOVEC} + g[(\rho Z)_{H_2O} + (\rho Z)_{NOVEC}] \quad (7)$$

As in the previous section, the evolution of the different temperatures collected during the experiments are analysed. Fig. 6 shows the temperatures for the experiments with 100, 200 and 300 W generated in the heat source. The most important conclusion is that the temperature of the heat source is notably lower than that in the previous experiments conducted at atmospheric pressure. A decrease of more than 10 °C is observed in all cases. This is a consequence of the lower evaporation temperature since the heat transfer increases abruptly when evaporation starts. This fact causes a higher heat transfer at lower temperatures of the

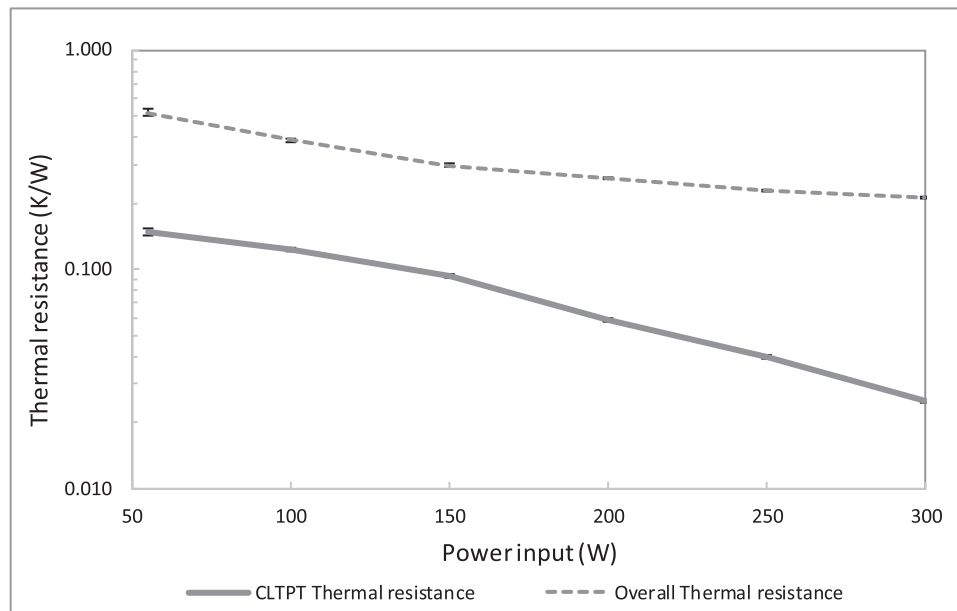


Fig. 5. Thermal resistance of the thermosiphon and overall system thermal resistance for atmospheric pressure initialization.

heat source. Another interesting difference from the previous experiments is that the temperatures of NOVEC and water virtually overlap. This means that the heat transfer between both fluids is very high. This is another consequence of the lower evaporation temperature since a higher flux of vapour of NOVEC is produced and these NOVEC bubbles ascend to reach the water region, as explained in Section 2. This process causes an aggressive heat exchange between both fluids; therefore, their temperatures are nearly the same. Regardless, the temperature of water is very similar to that of the previous experiments. This could be expected since the heat transfer conditions between water and the environment are the same.

The temperature difference between the aluminium block and NOVEC is nearly the same as in the experiments at atmospheric pressure and virtually constant throughout the experiment. The temperature difference between water and air is slightly lower than that in the tests at atmospheric pressure. In these experiments, the heating rates are also imposed by the thermal inertia of water. The total duration of the experiments is a few minutes shorter than those at atmospheric pressure since the delay of the water heating start is nearly non-existent. This is because the low pressure produces an immediate evaporation of NOVEC at the beginning of the heating process.

Again, the temperature of the heat source (aluminium block) is shown for all experiments since its cooling is the system goal. Fig. 7 shows the evolution of the temperatures for the experiments. The trend is similar to that of the atmospheric pressure experiments. Nevertheless, the stabilization temperature is clearly lower. This means that the system has a greater effectiveness when it operates at low pressures.

The main difference from the experiments at atmospheric pressure is the lower temperatures of the heat source and small temperature differences between NOVEC and water, while the difference between the water and the environment temperatures is similar to that of the previous experiments. In conclusion, removing the air from the condenser vessel interior produces an important increase in the cooling capacity of the system, and the heat source temperature can be decreased by 12–15 °C for the whole range of powers tested. This occurs since the lower pressure vapour has a higher specific volume, which creates larger bubbles, and the buoy-

ancy effect in the water column produces a higher rising stream and a higher stirring effect that homogenizes the temperature in the whole vessel.

Fig. 8 shows the thermal resistances of the thermosiphon and the overall systems, calculated as explained above (Fig. 5). In this case, both thermal resistances are lower than those calculated for the system at atmospheric pressure, but the thermosiphon thermal resistance is especially low. For power inputs higher than 100 W, the value is of the order of 0.001, which means a highly homogeneous temperature throughout the fluid region. This means that the thermosiphon works with a high fluid circulation, the heat exchange between water and NOVEC is efficient and the fluids in the condenser are highly agitated. This shows that the two-fluid condenser design has efficient behaviour.

In this case, the thermal resistance also decreases with power input due to the higher vapour mass flow rate. In this case, the decrease is more pronounced. The larger bubbles homogenize the fluids, which reduces the temperature differences that lead to low values of the thermal resistance.

3.3. Heat transfer analysis in the condenser

As one of the main novelties proposed in the present paper is the condenser charged with the mixture of NOVEC and water to improve the heat transfer performance, a set of tests were performed to analyse its effect in comparison to a condenser charged with NOVEC as the only coolant. To calculate the heat transfer coefficients, the setup was prepared as shown in Fig. 9. The condenser was insulated on the top and the base. The heat source and the hoses are also insulated to ensure that the heat generated in the heat source is transferred through the cylindrical finned lateral surface. Several thermocouples were located at three different heights inside the condenser to monitor the fluid temperature and on the cylindrical surface at the same heights as the internal ones to monitor the wall temperatures. Then, six tests were performed with the system charged with NOVEC, only filled up to the bottom of the condenser, and water in the rest of the condenser volume. Three tests were performed at atmospheric pressure and three other tests were conducted at 0.1 bar above the NOVEC saturation pressure. These trials were performed by generating 100, 200 and 300 W in the heat source. The same six tests

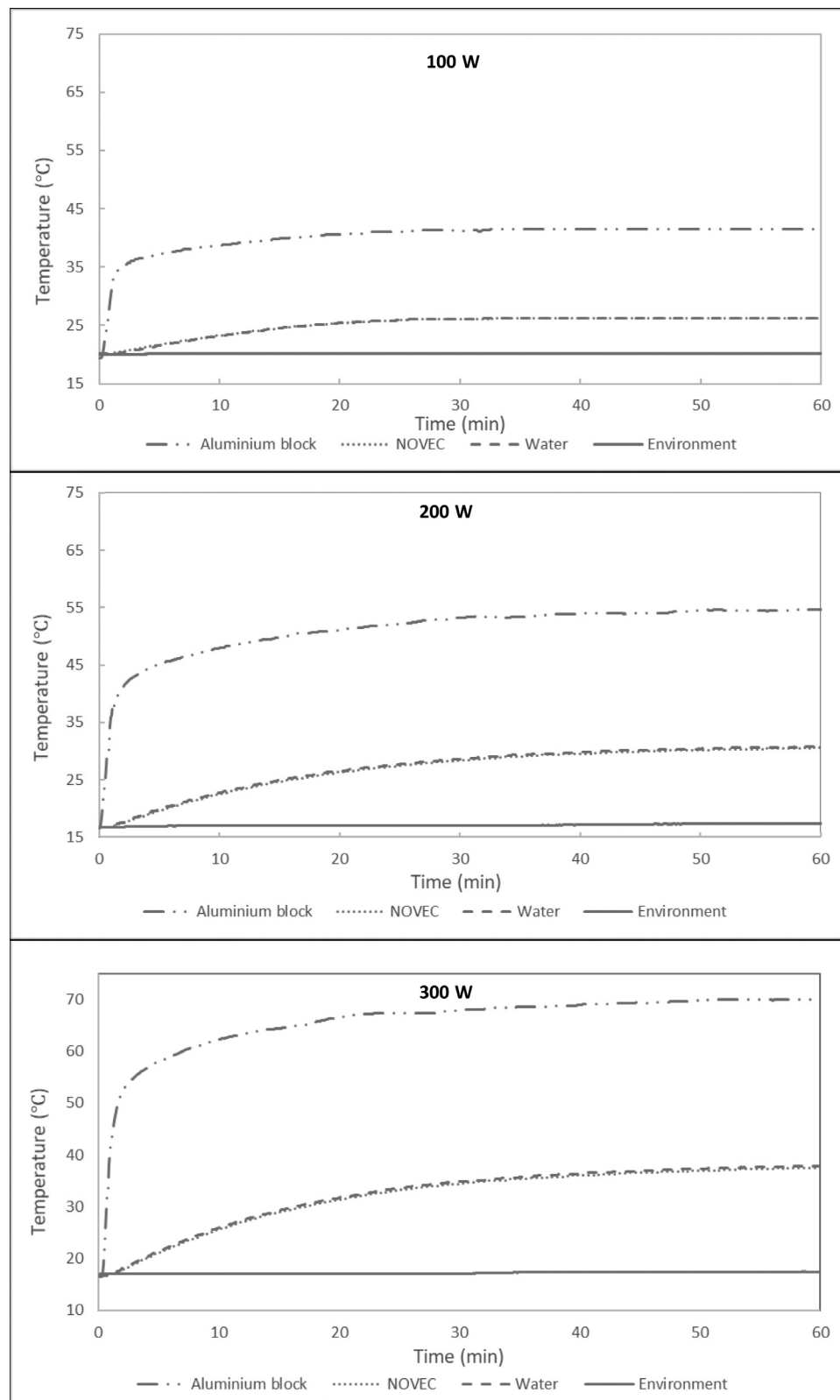


Fig. 6. Temperatures of the heat source, NOVEC fluid, water, and environment for experiments with 100, 200 and 300 W in the heat source and vacuum initialization.

(same power inputs and pressure conditions) were conducted with the condenser charged with the same amount of NOVEC but without water.

In some of the tests, the temperature inside the condenser is highly stratified, especially those with only NOVEC and atmo-

spheric pressure. In those cases, an average of the temperature difference was used to calculate a global HTC. Fig. 10 shows the HTC of all the tests and their uncertainties. The main result is the clear effect of the water and NOVEC combination that provides higher HTC than the tests performed without water. This

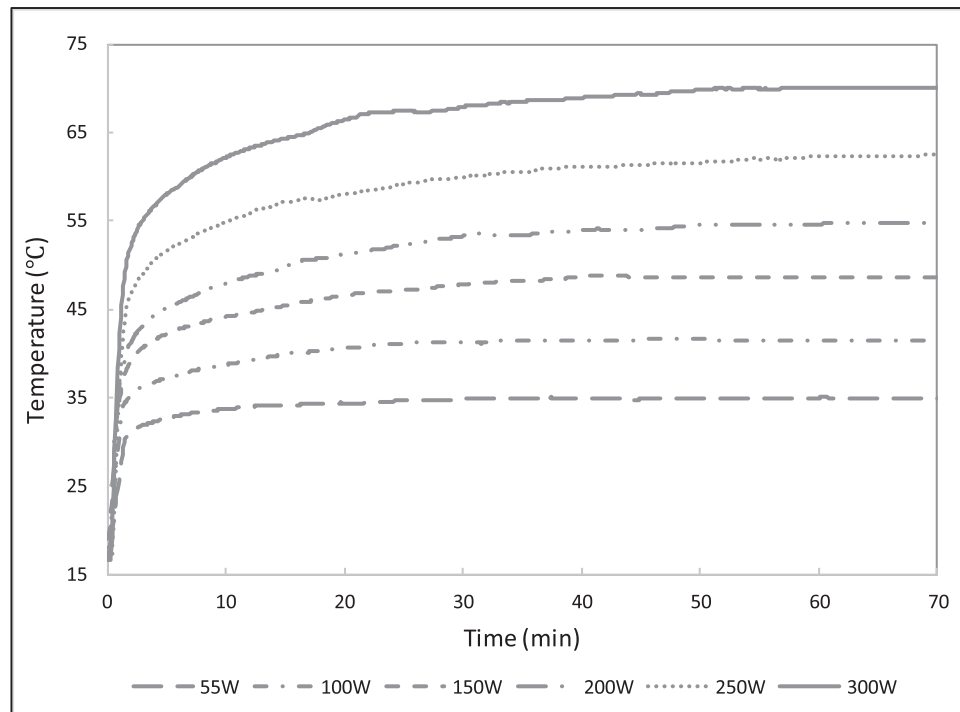


Fig. 7. Temperatures of the heat source for different heat source values from 55 W to 300 W and vacuum initialization.

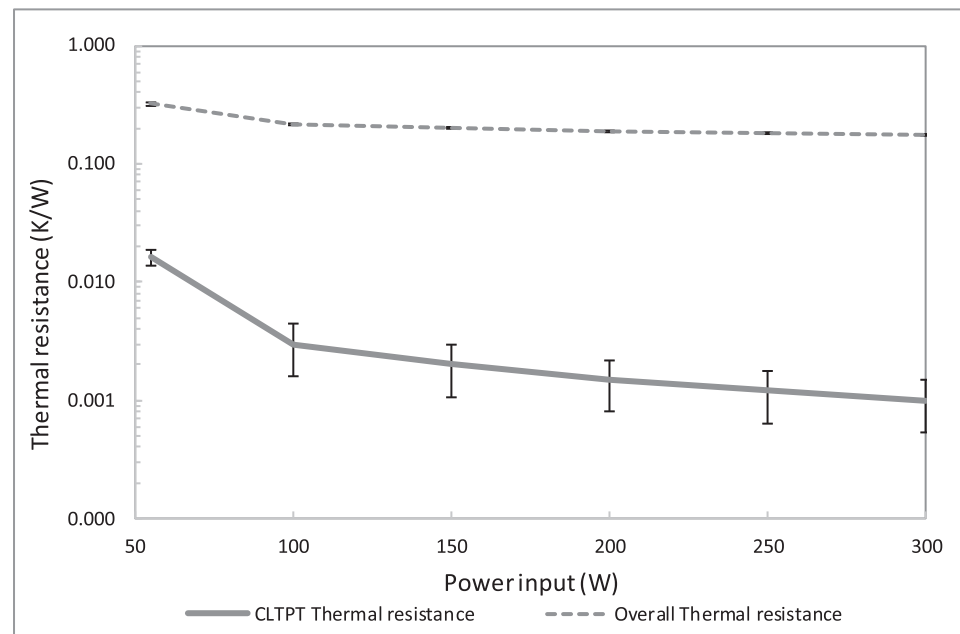


Fig. 8. Thermal resistance of the thermosiphon and overall thermal resistance for the system with vacuum initialization.

relates to the positive effect of the water, whose higher thermal conductivity increases the heat transfer to the walls in comparison to the vapour of NOVEC. The effect of vacuum is also noticeable in both cases (NOVEC + water and only NOVEC). It is known that the presence of non-condensable gases has a negative effect on condensation performance mainly due to the difficulty of diffusing the vapour through the gas mixture. In addition, a lower pressure decreases the evaporation temperature, which produces a higher vapour quality. As Honda et al. [78] demonstrated, the

vapour quality increases the condensation HTC. In the tests of the NOVEC and water mixture, the lower pressure also increases the heat transfer since it produces a more developed nucleate boiling regime and a higher vapour flux that stirs the water in the condenser and increases the heat transfer to the walls. In all cases, a slug regime in the raiser was observed, which means low vapour quality in the evaporator [79]. This regime produces an effective bubble lifting that improves the stirring of fluid in the condenser. Finally, the heat flux also produces an increase in HTC in all cases.

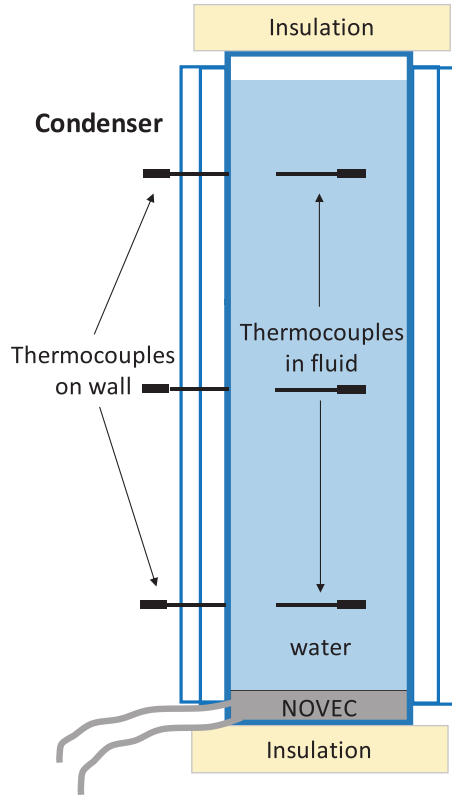


Fig. 9. Arrangement of the condenser to calculate the HTC.

The improvement in HTC observed is 133% for the case of lower heat flux at atmospheric pressure and in the range of 22–42% in the rest of the cases, which are significant increases in all cases.

The effect of the water produces a better homogenization of the temperatures in the condenser, which is positive since it reduces the temperature of the fluid in the return hose and consequently the heat source temperature, which is the main goal in the cooling of electronic devices. Nevertheless, the effect of HTC reduction could be noticeably higher if the external HTC of the condenser (heatsink to environment) was higher. As the system studied in the present paper is a completely passive setup, the external HTC is low since it is produced by the air of the environment, which is the main obstacle in the heat exchange path. The system may have a higher cooling performance increase in the case of an evaporative condenser that would increase the heatsink HTC and, indeed, the temperature difference between the condenser wall and the internal fluid.

4. Dynamic model for temperature prediction

In this section, a numerical model to calculate the temperatures of the different parts of the system is proposed. As a limited number of experiments were performed, the model allows extrapolating the results to other conditions or simulating similar systems with different geometrical parameters or even other fluids. The proposed model is based on heat transfer balances and some additional considerations.

Four elements are considered to represent the system. These elements are the aluminium block (1), the liquid NVEC (2), the water (3), and the heatsink (4). The temperatures of all elements are registered for each instant (time step) calculated from the beginning of the experiment. A heat transfer balance is stated for each element at every time step, and the evolution of temperatures is calculated.

The model is formulated under some hypothesis simplifying the complex heat and mass transport processes that occur in the fluids. The first hypothesis is that of homogeneous temperatures in each fluid, the aluminium block and the heatsink. The homogeneity of the fluids was checked in some previous experiments with several thermocouples inside the vessel. This is a result of the fact that the evaporation process produces a high stir and turbulence in both fluids, NOVEC and water. On the other hand, the aluminium block is a small and thin piece with a high thermal conductivity. Therefore, its temperature can be considered homogeneous. In a similar way, the condenser heatsink also has a high thermal conductivity, and the heat transfer is predominantly radial. The exceptions for the heatsink are the fins, whose temperature is expected to be lower than that of the vessel. Therefore, the heat transfer efficiency of the fins must be considered in the external HTC. Another hypothesis is related to the fact that the NOVEC bubbles reach the water region in a short time from their generation. Therefore, it is assumed that a part of the heat transferred from the aluminium block, represented by the parameter λ , is directly transported to the water. The mass of every element is considered constant since the evaporated fractions of NOVEC or water are quickly condensed and the differences are negligible. The heat transfer between the four elements is represented in Fig. 11.

For each element (i), energy conservation must be fulfilled; therefore, the difference between the incoming and outgoing heats produces a heat increase (Eq. (8)). The heat transferred between elements is calculated as a function of temperature difference and a coefficient “h”, which represents the global HTC between elements (Eq. (9)). This means that it can be a convective coefficient in the case of the water and the heatsink, coefficient of contact between fluids (NOVEC and water) or a compound wall coefficient from the aluminium block to NOVEC through the NOVEC evaporator capsule wall.

The heat transferred by the gas NOVEC needs a different treatment since it is generated in contact with the capsule wall and ascends through the NOVEC liquid until reaching the water, where the bubbles transfer their heat and condense, as shown in Fig. 2. To model this process in a simple way, a coefficient “ λ ” is used to quantify the fraction of heat from the aluminium block that reaches the water through the bubbles. This parameter follows the formulation shown in Eq. (10). It is a function of the initial temperature T^0 , the temperature of liquid NOVEC T_2 and the evaporation temperature of NOVEC, which follows Eq. (6). With this formulation, when the liquid NOVEC is closer to the evaporation point, λ is closer to 1, which means that more gas reaches the water.

The fourth heat transfer equation can be combined into a linear equation system (11) whose coefficients are known. In this way, the temperatures of the four elements can be solved together in a simple operation. The coefficients of equation system (11) are shown in Table 3. The values of the parameters used in the modelling equations are shown in Table 4. These values were measured or adjusted (ad) for the model. In addition, the operating conditions and material main parameters used in the model are shown in Table 5.

$$(\dot{Q}_{in_i} - \dot{Q}_{out_i}) = \frac{m_i c_{p_i} (T_i - T_i^{t-\Delta t})}{\Delta t} \quad (8)$$

$$\dot{Q}_{out_i} = \dot{Q}_{in_{i+1}} = h_{i,i+1} A_{i,i+1} (T_i - T_{i+1}) \quad (9)$$

$$\lambda = K \frac{T_2 - T^0}{T_{evap \text{ NOVEC}} - T^0} \quad (10)$$

$$\begin{pmatrix} a_{1,1} & a_{1,2} & 0 & 0 \\ a_{2,1} & a_{2,2} & a_{2,3} & 0 \\ a_{3,1} & a_{3,2} & a_{3,3} & a_{3,4} \\ 0 & 0 & a_{3,4} & a_{4,4} \end{pmatrix} \begin{pmatrix} T_1 \\ T_2 \\ T_3 \\ T_4 \end{pmatrix} = \begin{pmatrix} \beta_1 \\ \beta_2 \\ \beta_3 \\ \beta_4 \end{pmatrix} \quad (11)$$

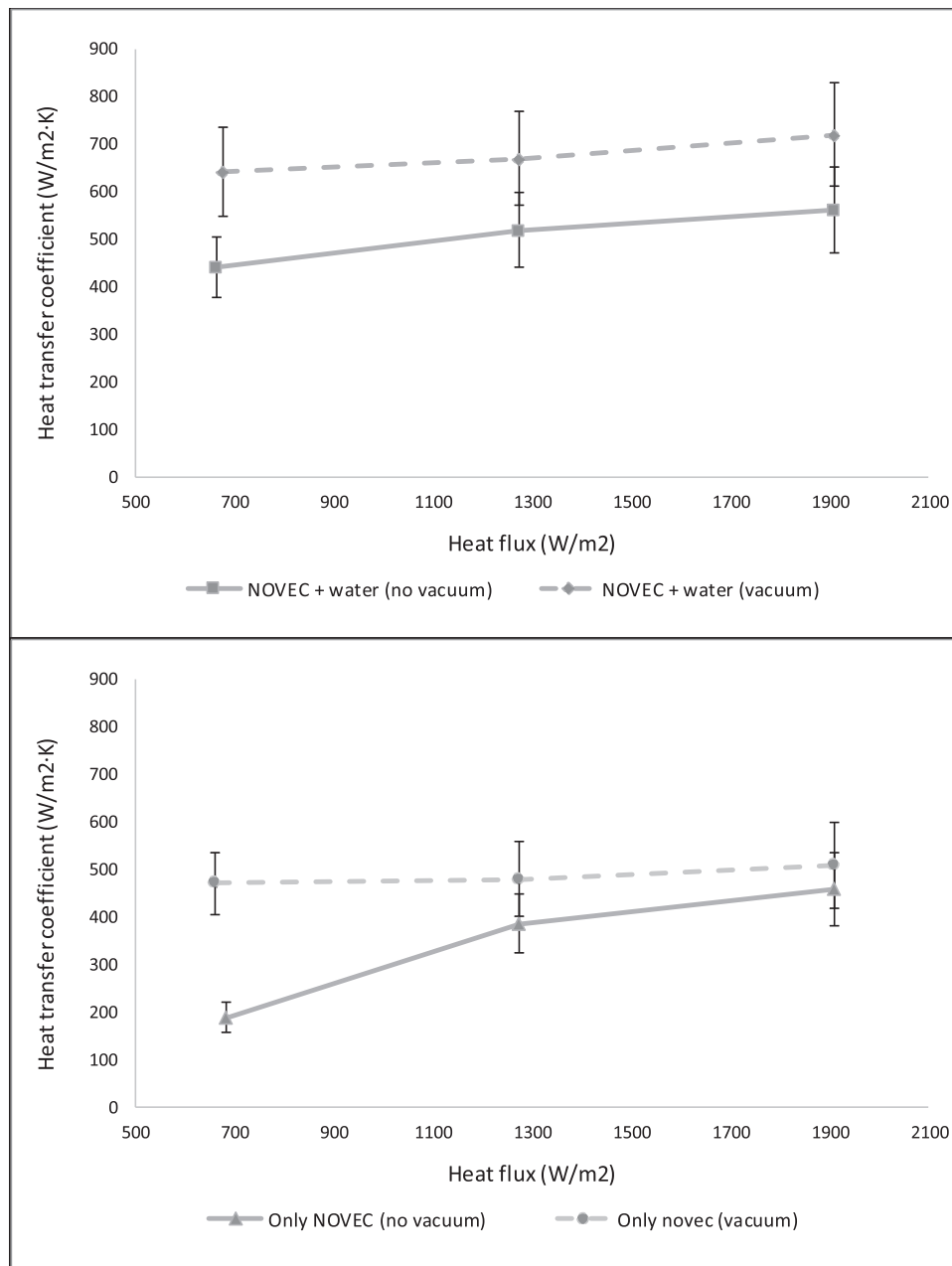


Fig. 10. HTC for the combination of NOVEC + water (above) and only NOVEC (below) inside the condenser.

Table 3

Coefficients of the linear equation system (11).

Element (i)	$a_{i,1}$	$a_{i,2}$	$a_{i,3}$	$a_{i,4}$	β_i
Block (1)	$-\frac{m_1 c_{p1}}{\Delta t} - h_{1,2} A_{1,2}$	$h_{1,2} A_{1,2}$	0	0	$-\frac{m_1 c_{p1} T_1^{i-\Delta t}}{\Delta t} - Q_{gen}$
NOVEC (2)	$(1 - \lambda) \cdot h_{1,2} A_{1,2}$	$-\frac{m_2 c_{p2}}{\Delta t} - (1 - \lambda) \cdot h_{1,2} A_{1,2} - h_{2,3} A_{2,3}$	$h_{2,3} A_{2,3}$	0	$-\frac{m_2 c_{p2} T_2^{i-\Delta t}}{\Delta t}$
Water (3)	$\lambda \cdot h_{1,2} A_{1,2}$	$-\lambda \cdot h_{1,2} A_{1,2} + h_{2,3} A_{2,3}$	$-\frac{m_3 c_{p3}}{\Delta t} - h_{2,3} A_{2,3} - h_{3,4} A_{3,4}$	$h_{3,4} A_{3,4}$	$-\frac{m_3 c_{p3} T_3^{i-\Delta t}}{\Delta t}$
Heatsink (4)	0	0	$h_{3,4} A_{3,4}$	$-\frac{m_4 c_{p4}}{\Delta t} - h_{3,4} A_{3,4} - h_{ext} A_{ext}$	$-\frac{m_4 c_{p4} T_4^{i-\Delta t}}{\Delta t} - h_{ext} A_{ext} T_{ext}$

Table 4

Values of parameters used in the model.

Parameter	1,2	2,3	3,4	Exterior
$A_{i,j}$ (m²)	0.00138	0.00921	0.238	1.27
$h_{i,j}$ (W/m²·K)	$4016 + 9.16 \cdot Q_{gen}$ (ad)	200 (ad)	550	12.1
K (-)	$1.095 - 0.0022 \cdot Q_{gen}$ (ad)			
T^0 (°C)	18 (ad)			

5. Model results, application range and limitations

The model was applied to the conditions of all the experiments performed. The pressure conditions, atmospheric conditions and vacuum conditions at the initialization and the whole range of values of power generation in the heat source were tested. The accuracy of the predictions depends on the coefficients used in the calculations. The correct prediction of the temperature of the alu-

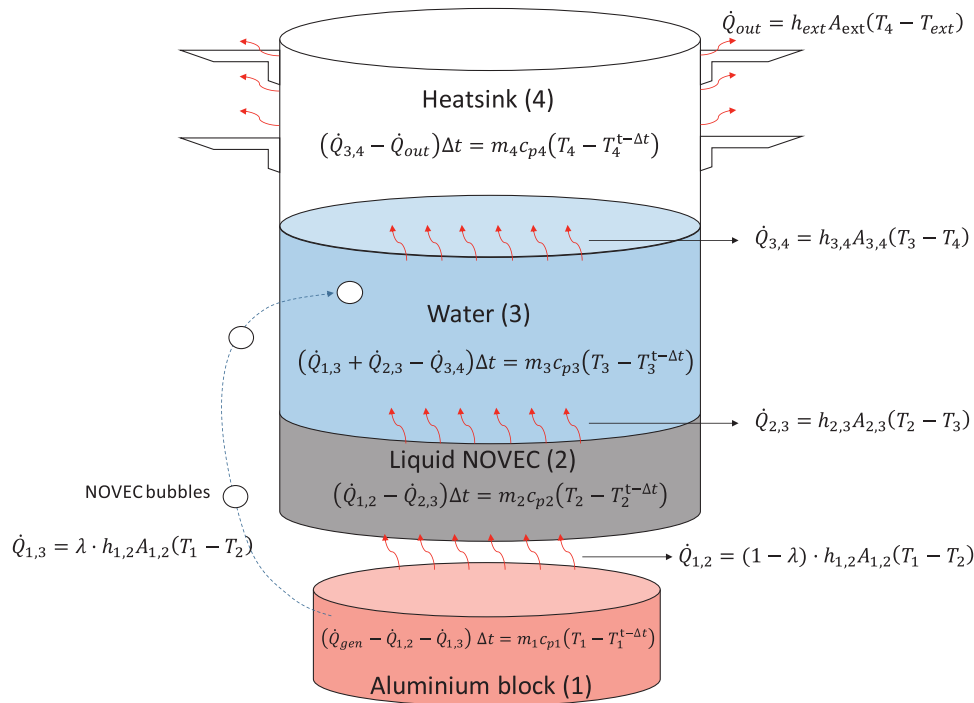


Fig. 11. Scheme of the heat balances applied to the model for temperature prediction.

Table 5
Operating conditions and main material parameters used in the model.

Parameter	Heat source	NOVEC	Water	Vessel
Mass (kg)	0.2	0.5	3	2.5
Specific heat (J/kg.K)	897	See Table 1	4180	385
Initial temperature (°C)	17–20.5 (depending on experiments)			
Time step size (s)	0.1			
Pressure (bar)	Eq. (8) ($P_{air} = 1.01$ for atmospheric tests, $P_{air} = 0.1$ for vacuum tests)			
Power inputs (W)	55–300 (depending on experiments)			

minium block depends on the coefficient $h_{1,2}$. This coefficient is adjusted through a linear function of the heat source, and it can be improved with more accurate functions and extended to other heat source ranges. In a similar way, the prediction of the NOVEC temperature depends on the coefficient K of Eq. (10). To predict reasonable results in the whole range of power inputs, the parameters K and h were adjusted by using the experimental data.

Figs. 12,13 show the comparison between the predicted and experimental temperatures of the aluminium block, NOVEC and water for the experiments initialized with atmospheric pressure and vacuum, respectively. In both cases, the illustrated results correspond to values of 100, 200 and 300 W in the heat source.

In general, the model predictions show reasonably good behaviour in the heating and stabilization stages for all the elements. The predictions of the stabilization temperatures were accurate, especially in the case of vacuum initialization, in which the error was lower than 1 °C. However, a maximum error of 5 °C is found in the test of 100 W at atmospheric pressure for the aluminium block temperature. The difference in temperatures between NOVEC and water, which is significant in the atmospheric experiments and very small in the vacuum tests, is also reasonably well predicted by the model. This difference is more pronounced for lower heat sources and is reduced for higher values since the evaporation of NOVEC is more aggressive. The different behaviour in the vacuum tests is well represented by the coefficient λ since it depends on

the NOVEC evaporation temperature, which is highly decreased in the vacuum tests.

In the tests with vacuum initialization, the predictions are highly accurate. Both stages, heating and stabilization, are reasonably well predicted for the three thermocouples tested. In this case, the system has a simpler behaviour since the heat transfer from NOVEC to water is high due to the quick and intense beginning of evaporation, and the experimental temperatures of NOVEC and water are virtually the same. This fact is well represented by the model except for at the beginning of the experiment, when the NOVEC heating suffers a slight delay in the experimental data. This may be caused by the thermal inertia of the hoses and some heat losses from the hoses to the surroundings that were not considered in the model.

Overall, this model can be considered a good approximation for representing the system behaviour. The transient evolution of temperatures is reasonably well represented, and the stabilization values are reasonably close to the experimental data considering the simplicity of the model and the complex processes that occur inside the system.

This numerical model is based on a linear equation system in which most of the coefficients are not temperature dependent. On the one hand, some of those coefficients are temperature dependent since they depend on the specific heat and densities of NOVEC. The properties of NOVEC are shown in Table 1. Nevertheless, the variation of these properties in the range of temper-

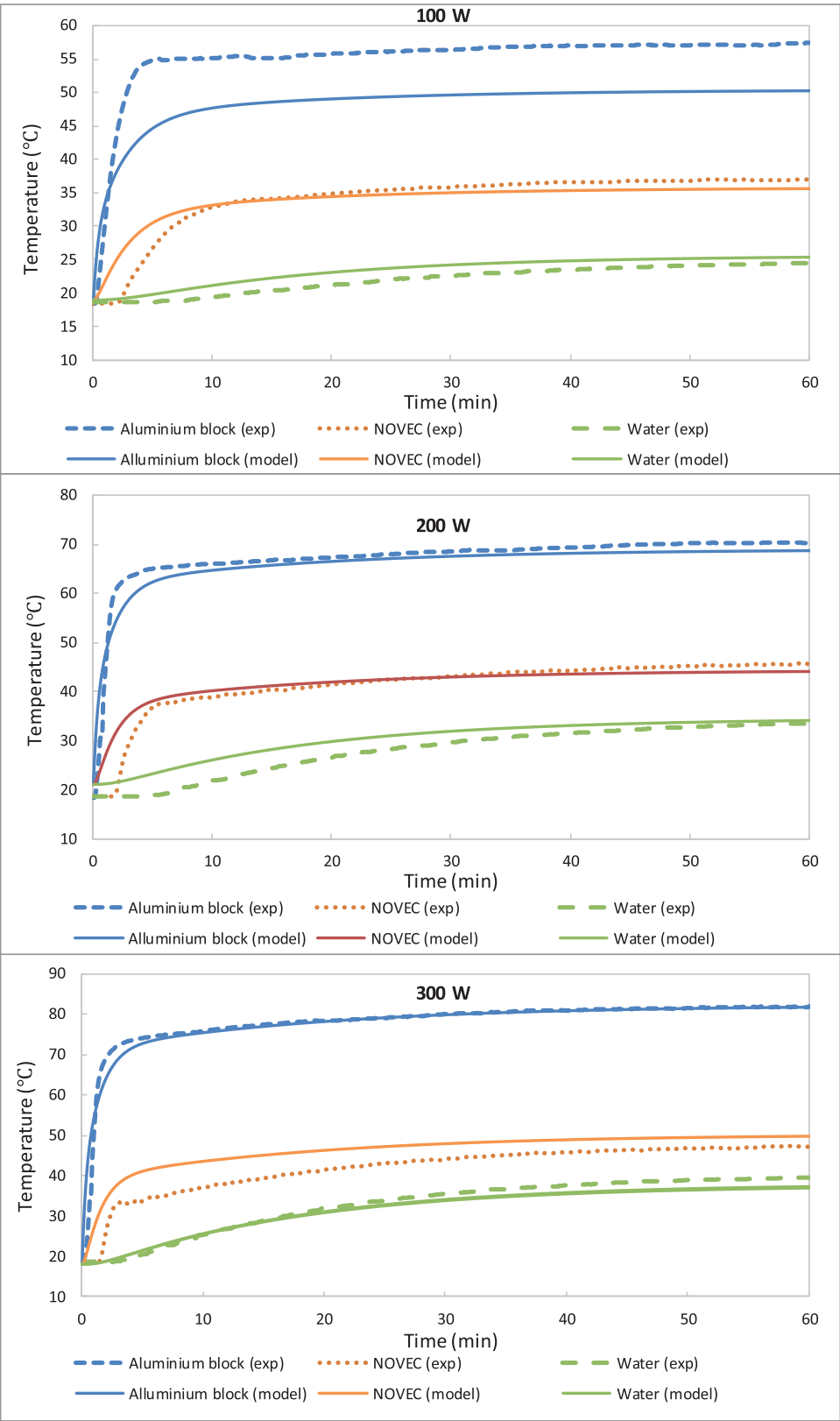


Fig. 12. Comparison of predicted and experimental temperatures for the tests with 100 W, 200 W and 300 W in the heat source and atmospheric pressure initialization.

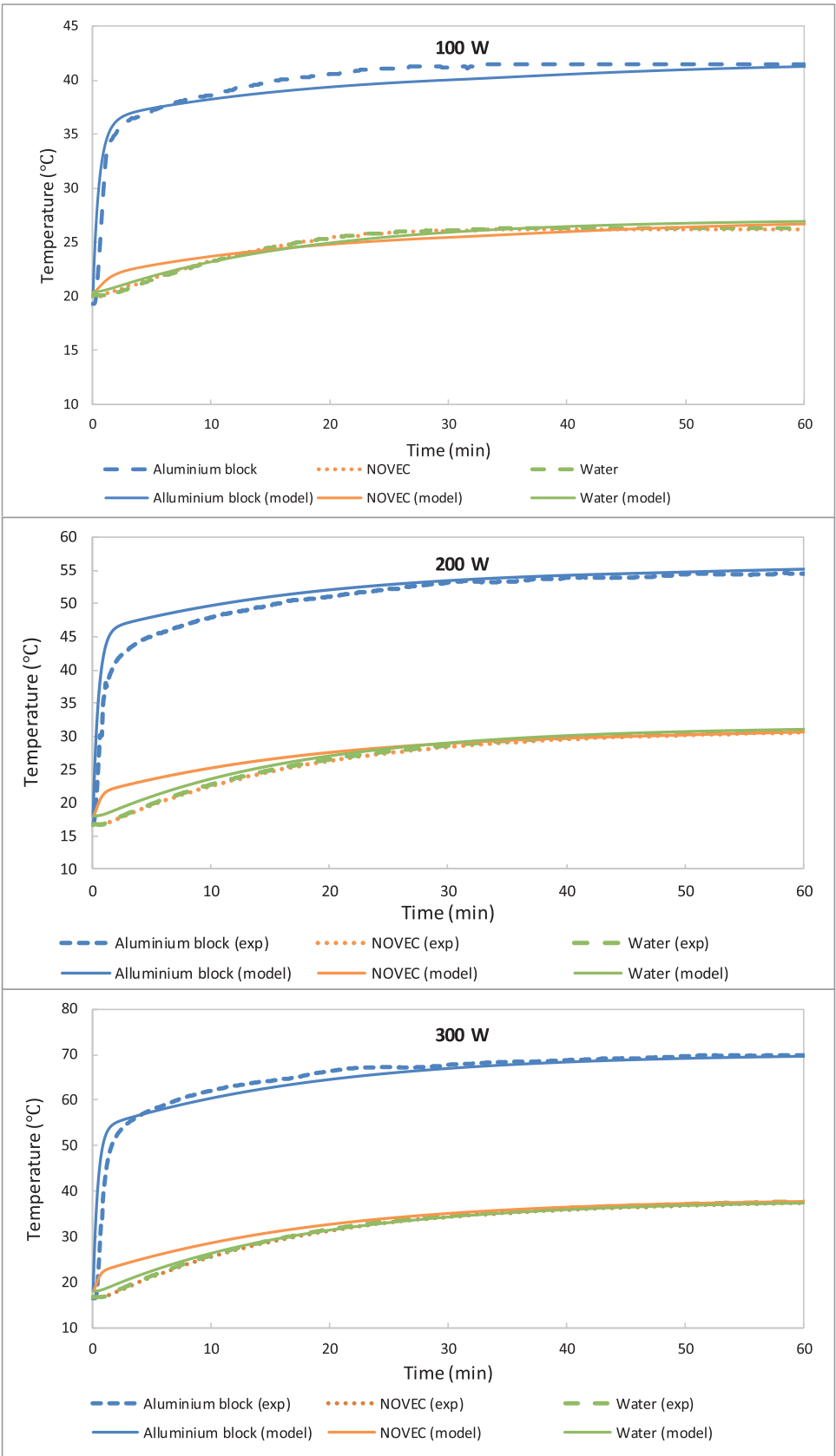


Fig. 13. Comparison of predicted and experimental temperatures for the tests with 100 W, 200 W and 300 W in the heat source and vacuum initialization.

atures at which the system operates is low. On the other hand, the parameter λ (Eq. (10)) is also dependent on temperature. To address this fact, the equation system can be solved through an explicit time discretization scheme by solving these temperature-dependent parameters as functions of the temperature of the previous instant. This method is not accurate for large time steps. Another option is to apply an implicit scheme, but this needs an iterative process to converge each time step. In this case, as the equation system has only four temperatures to solve and the temperature-dependent parameters follow simple equations, the convergence of each time step only requires a few iterations. Therefore, the implicit scheme was used for the calculation with a time step size of 0.1 s. All the calculations performed with this model were solved in only a few seconds, which shows the low computational effort of the model.

The goal of such a model is to predict the system behaviour in the range of power and boundary conditions tested. This allows predicting the system thermal behaviour for different operating conditions, geometrical dimensions or fluid mass quantities, which makes it a useful tool for optimizing the design of this kind of cooling system. The application to a wider range of boundary conditions may be possible after a new adjustment of the model parameters, which would require new tests.

As the model follows a simple formulation, based on spatial and temporal first-order schemes, in which the four system main elements are considered four finite volumes, the model is only applicable if two conditions are fulfilled. First, the solid elements (heat source and heatsink) must be thermally thin elements. This means that their Biot number < 0.1 . Second, the fluids (NOVEC and water) must be highly stirred. This occurs when the evaporator operates in a fully developed nucleate boiling regime. Both conditions are fulfilled in all the experiments performed in this work.

6. Conclusions

This paper presents and tests a novel cooling system for electronic components based on a CLTPT with an immiscible combination of NOVEC 7000 and water in the condenser that transports heat from a solid hot source to a heatsink without any energy expenditure. The system uses the low evaporation temperature of NOVEC to increase the cooling efficiency and buoyancy flow of vapour to produce quick fluid movement. The condenser is based on a water pool in which the NOVEC bubbles are efficiently condensed without the effect of non-condensable gases and the water suffers an important agitation that increases the heat transfer to the walls.

A series of experimental tests with different power values in the heat source were carried out to assess the cooling performance of the system under different conditions. Likewise, the system was tested under different starting pressure conditions, namely, at atmospheric pressure and in vacuum.

The tests showed good cooling behaviour of the system since the heat source was kept at a relatively low temperature for the whole range of the tested heat source power values. Temperatures lower than 70 °C were kept for a heat flux of 220 kW/m² in the heat source. The pressure reduction inside the system notably reduced the temperature of the heat source, which is the eventual target of the system design. The system also showed a low thermal resistance into the thermosiphon, which relates to the good fluid circulation and agitation inside the condenser. The effect of water in the condenser was also tested. The results showed an increase in the HTC of 22% to 133% with respect to the same tests without water.

The system operated for all conditions automatically with continuous flow circulation without any external pumping action, which is an advantage in comparison with other liquid cooling sys-

tems. In addition, only a small amount of NOVEC was needed for correct operation of the system, and no fluid losses were produced since the system was tightly closed. These facts reduce the system costs and enable its application to electronic systems that require high cooling reliability at a reasonable cost.

In addition to the experimental tests, a numerical model was proposed, which was shown to predict the experimental results reasonably well, allowing the study of the system behaviour for different external or system conditions.

The main limitation of the system is the dependence on the heatsink surface and external convection conditions. In this respect, the whole system may increase its performance if the heatsink operates as an evaporative condenser. The study also presents some limitations that could be addressed in future works. These limitations are mainly the lack of some measurements, such as a flowmeter, to evaluate the mass flow and assess the vapour quality in the riser. Another area to improve may be the use of an evaporator with a geometry suitable to test the heat transfer coefficients in the evaporator.

Declaration of Competing Interest

Funding for open access charge: Universidade de Vigo/CISUG. The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

CRediT authorship contribution statement

M.A. Gómez: Conceptualization, Methodology, Investigation, Writing – original draft, Writing – review & editing, Supervision. **R. Bellas:** Conceptualization, Methodology, Investigation, Writing – original draft, resources, Supervision. **A. González-Gil:** Visualization, Investigation, Writing – original draft. **A. Cacabelos:** Methodology, Visualization, resources. **A. Larrañaga:** Investigation, Software, Formal analysis.

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