

Article

Prototyping and Testing of M-Cycle Indirect Regenerative Evaporative Cooling System for Electronics Cooling

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Abstract

This paper presents the design and experimental investigation of an indirect regenerative evaporative cooling (IREC) system based on the M-cycle concept for electronic systems operating in high ambient temperature. The proposed heat exchanger consists of dry and wet vertical channels separated by thin copper walls and equipped with an open-water reservoir located at the bottom of the cooler. Airflow is introduced into the dry channels, where it is pre-cooled and directed toward a heat source. Next, it is redirected into the wet channels in a counter-flow arrangement. The heat source consists of power MOSFET transistors operating in current-source mode, with enhanced convective heat transfer achieved through attached heat sinks. Temperature and relative humidity sensors were installed at key measurement points to monitor thermodynamic conditions in the exchanger. The findings demonstrate the potential of the IREC approach as an effective and energy-efficient solution for cooling electronic systems in elevated temperature environments.

Keywords: Maisotsenko cycle (M-cycle); indirect regenerative evaporative cooling (IREC); dual-phase heat exchanger; electronics cooling systems; psychrometric chart; capillary water absorption

1. Introduction

The miniaturization of electronic components, combined with increasing power densities in modern electronic systems, has made thermal management one of the most significant challenges in contemporary electronics engineering [1,2]. As a result, conventional air-based cooling solutions have become insufficient, driving the research of more effective and energy-efficient alternatives [3,4]. Inadequate heat dissipation leads directly to elevated junction temperatures, accelerated component degradation, and reduced system reliability, making the development of advanced cooling strategies essential for both system performance and lifespan of electronic components [5,6].

Among the broad spectrum of cooling technologies investigated in recent years, two-phase evaporative cooling systems have attracted considerable attention due to their su-

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perior heat transfer characteristics compared to single-phase approaches [7,8]. By exploiting the latent heat of vaporization of the working fluid, two-phase systems achieve significantly higher effective heat transfer coefficients, enabling the removal of high heat fluxes at relatively low and uniform surface temperatures [9,10]. Pump-driven two-phase loops, in which the working fluid is actively circulated through evaporator and condenser sections, represent a particularly practical implementation of this concept, offering controllability and scalability for electronics cooling applications [8,11,12]. Various configurations have been studied, from on-chip cooling of individual processors to rack- and cabinet-level systems for data centers [3,10], demonstrating the versatility and maturity of the approach.

Evaporative cooling technologies based on the Maisotsenko cycle (M-cycle) are a promising solution for energy-efficient thermal management in electronics [13,14]. The M-cycle is based on a thermodynamic process in which the working air stream is pre-cooled in dry channels before being used to evaporate water in adjacent wet channels. Thanks to this, the temperature at the outlet of the dry channel used for cooling the electronic system, is close to the dew point temperature [15–17]. This fundamental thermodynamic advantage leads to an increase in cooling potential significantly exceeding that of standard evaporative coolers, while consuming only a fraction of the energy required by vapor-compression refrigeration systems [13,14]. The M-cycle concept has been extensively studied numerically and experimentally for HVAC applications, with research covering heat exchanger geometry optimization [18,19], integration with desiccant systems [20,21], the influence of airflow distribution [22], wetting mechanisms [23,24], and membrane materials [25]. The feasibility of miniaturizing the M-cycle exchangers for electronics cooling purposes was demonstrated experimentally by Sadighi Dizaji et al. [26], and more recently a full prototype of an indirect regenerative evaporative cooling system targeting electronics applications was reported by Levchenko et al. [27], confirming that M-cycle-based cooling is not limited to building HVAC applications.

Despite the individual maturity of both the two-phase solution for cooling and M-cycle regenerative evaporative cooling, their combined use in a single integrated architecture for electronics cooling has received limited attention. The M-cycle cooling approach provides pre-cooling of the heat rejection medium, which has the potential to combine the high heat flux capacity of cooling systems with the energy efficiency of sub-wet-bulb cooling [12,27]. Such a configuration could prove particularly advantageous in environments where ambient temperatures are elevated or where the energy consumption of conventional refrigeration-based rejection systems is undesirable.

The present work addresses this gap by describing the simple design, prototyping, and experimental characterization of an M-cycle cooling system intended for electronics applications. The system architecture, component selection as well as preliminary thermal performance results are presented and discussed. Although M-cycle technology has been widely studied in the context of air conditioning, its application to electronics cooling remains relatively underexplored. Recent experimental works have demonstrated the feasibility of downscaling M-cycle systems for thermal management of electronic components. These findings indicate that M-cycle-based cooling systems can provide an attractive alternative to conventional methods, especially for applications requiring energy efficiency, simplicity, and environmental sustainability. In this study, we experimentally tested the performance of the cooling system at high ambient temperature, a condition where conventional forced convection cooling systems fail, resulting in runaway performance in electronic devices.

2. Materials and Methods

A demonstrator of IREC/FFR (indirect regenerative evaporative cooling/Full Flow Return) technology was built for electronic system cooling, with an exposed water reservoir at the bottom beneath the exchanger (Figure 1). A simple exchanger design was used, with vertical channels: dry (DC) and wet (WC) ones with dimensions of $170 \times 45 \text{ mm}^2$. The channels are separated by $200 \mu\text{m}$ thick copper sheet. Air is introduced from the top into the dry channels, then moves horizontally to the heat source, where it returns to cool the electronic system. It then moves horizontally through the wet channels in the opposite direction (compared to the airflow in the dry channels) and is exhaled by a fan located at the outlet of the wet channels.

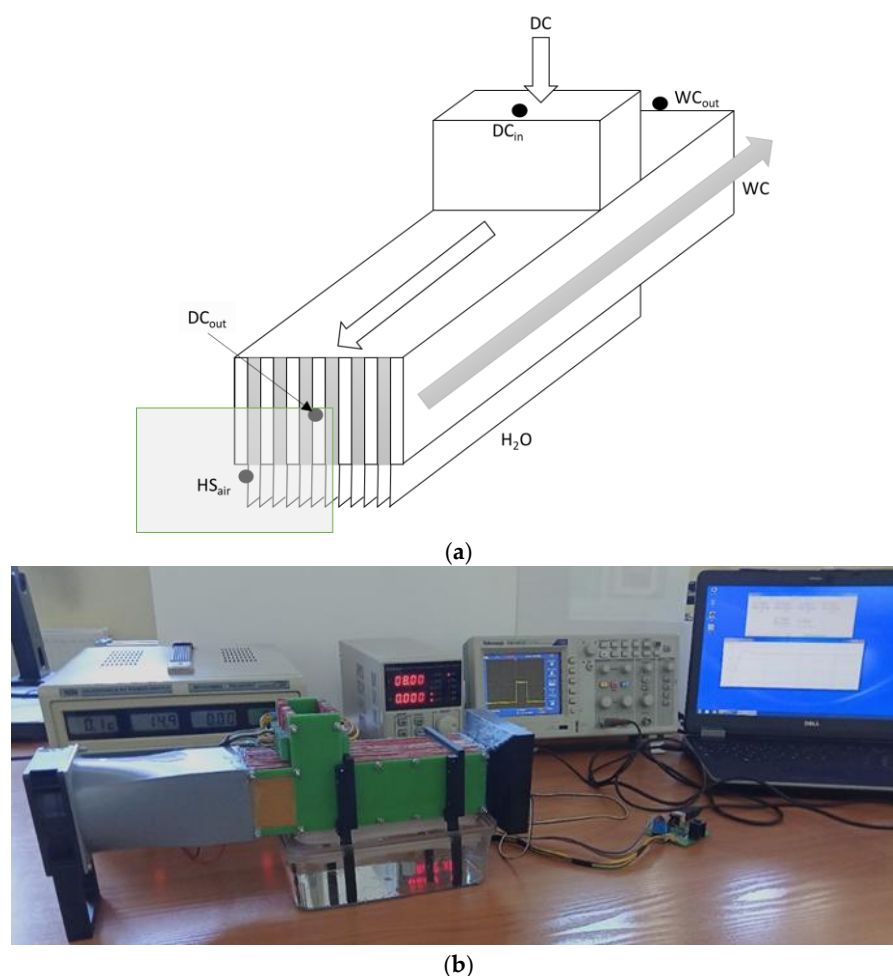


Figure 1. Schematic diagram of the IREC technology demonstrator with a lower water reservoir for heat dissipation in electronic systems: (a) schematic diagram with marked temperature and relative humidity sensors; (b) view of the demonstrator during testing.

On the wet side WC, a porous material made of bamboo fibers with a thickness of approximately $200 \mu\text{m}$ was attached using double-sided adhesive tape (Figure 2). On the dry side DC, copper terminals ($3 \times 5 \text{ mm}^2$) were soldered to introduce airflow turbulence and increase heat exchange between the DCs and WCs (Figure 3).

The copper walls of the DCs and WCs were soldered to frames manufactured using printed circuit board (PCB) technology. Temperature and relative humidity sensors were located on the frames and connected to the measurement system near the inlet. A total of four sensors were mounted in the air channels: one at the DC inlet, one at the DC outlet,

one at the wet channel (WC) outlet, and one at the heat source, where the airflow reverses (a 180° turn) towards the wet channel (Figure 1).

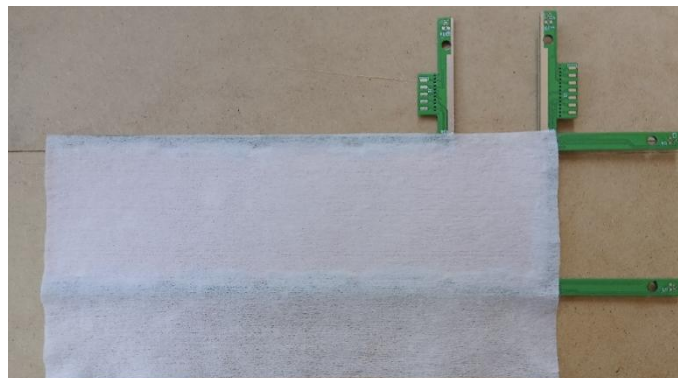


Figure 2. Porous thin bamboo material glued to the wall of a wet channel.

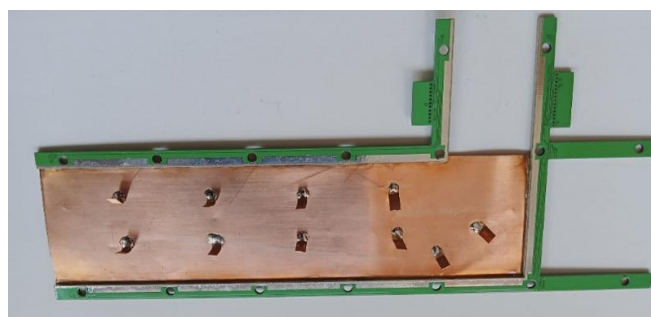


Figure 3. Dry channel wall with elements increasing the turbulence of the flowing air.

The heat source imitating the power electronic system was a set of three high-transconductance power MOSFET transistors operating in current source mode and powered by an external power supply with a voltage of $U = 8\text{ V}$ and a maximum current of $I_{max} = 5\text{ A}$. The transistors were soldered to a PCB that additionally featured a gate control circuit. Heat sinks ($14 \times 14\text{ mm}$, $h = 10\text{ mm}$, $R_{th} = 27,4\text{ K/W}$) were bonded to the top surface of the transistors with thermally conductive adhesive to enhance convective heat transfer during the flow of cool air generated in the DCs. A temperature and relative humidity sensor HS_{air} was placed next to the transistors to measure air parameters near the heating elements (Figure 4).

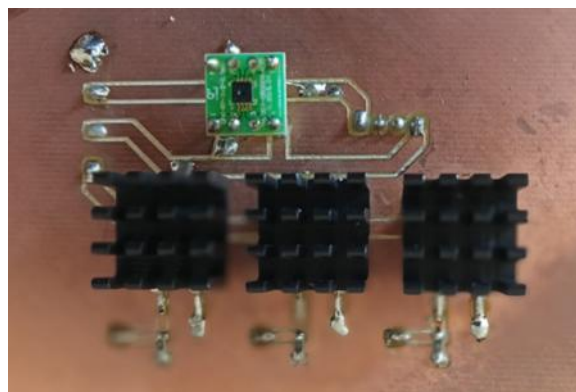


Figure 4. Heat source in the form of 3 MOSFET power transistors with heatsinks and temperature and relative temperature sensors HS_{air} .

The developed heat exchanger contained six wet channels (WCs) and seven dry channels (DCs). The entire structure was sealed at the edges and fastened with screws. Additionally, the mechanical connections between the channel structure, the exhaust system, and the heat source cooling chamber were sealed with silicone gel. A simple microprocessor controller was developed to control the power of the heat sources (transistors) and stabilize the temperature, as well as to transmit and record measurement results to a computer. Software was developed in the .NET environment and C# language for analysis, visualization and archiving of test results.

3. Experimental Results

Before assembling the heat exchanger, tests were conducted to assess the water absorption efficiency of the porous material. A thin bamboo fabric was glued to a thin copper sheet 100 mm high, leaving a portion of the fabric protruding beyond the metal on its longer edge (Figure 5). This section of fabric was immersed in water, and the water level in the fabric was checked after approximately 3 min. The result was satisfactory—the entire fabric was wet, despite the water being lifted upwards, against gravity. Next, the fabric was dried, and the capillary water absorption test was repeated. As expected, the entire material became wet again in less than 3 min. The results of water absorption and wetting of the porous material are shown in Figure 5. Note that in the developed heat exchanger, the height of the water column lifted by capillary forces is less than half as high, and amounts to 45 mm.

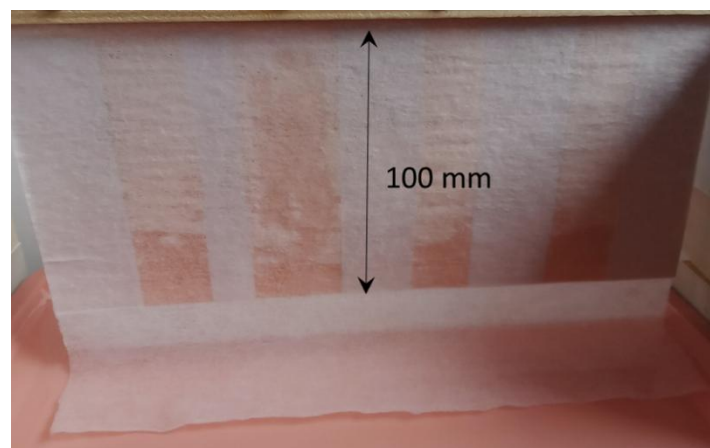


Figure 5. The effect of capillary rising of water in a porous material.

The next stage of the research involved measuring the heat source (HS) power, its temperature, and air parameters (temperature and relative humidity) at selected points of the IREC/FFR exchanger: at the inlet of the dry channel (DC_{in}), at the outlet of the dry channel (DC_{out}), at the outlet of the wet channel (WC_{out}), and around the heat source (HS_{air}) (Figure 1). A special air preparation chamber was constructed for the research, containing the heat exchanger and supplying air at various temperatures from a fan-equipped air-stream generator. The chamber had small openings at the top and bottom to maintain a near-atmospheric pressure inside. Despite this, the air pressure in the chamber increased slightly above the ambient pressure. The pressure increase was noticeable in the measurements taken by the sensor at the inlet of the dry channel (DC_{in}). The increase in specific humidity caused by the increase in air pressure in the air preparation chamber was corrected. The working exchanger equipped with a fan located in the chamber opening allowed for proper air circulation around the heat source. The mean value of fan supply power was equal to 1.8 W. The main goal of the research was to verify the operation of the exchanger in the M-cycle at high ambient temperatures. Thanks to the air preparation

chamber, the achieved air temperature around the heat exchanger was approximately 55 °C. Figure 1 schematically shows the sensor layout in the dry and wet channels. The design utilizes temperature and relative humidity sensors with a measurement accuracy of $\Delta RH < \pm 1.0\%$ RH and $\Delta T < \pm 0.1$ °C. Temperature and relative humidity measurement results were determined in a steady state, averaging $N = 100$ partial measurements.

As a result, during the experiments, the air temperature and relative humidity were measured at the dry channel inlet (T_{DCin} , RH_{DCin}), at the dry channel outlet (T_{DCout} , RH_{DCout}), at the wet channel outlet (T_{WCout} , RH_{WCout}), and at the heat source (T_{HSair} , RH_{HSair}). Additionally, the heat source temperature and power dissipated were measured.

The tests were conducted for three ambient air temperature values: approximately 28 °C, 41 °C, and 55 °C. These temperatures were achieved using the air-stream generator available in the experimental setup, and they correspond to the stable operating conditions that could be generated by this heater at different power levels. The thermal equilibrium was achieved after about 15 min of stabilization when the temperature fluctuation was lower than ± 0.2 °C. The achieved ambient conditions are realistic and can occur in real thermal management systems. Ambient temperatures higher than 55 °C are rarely encountered in practice.

Simultaneously, the heat source temperature was varied from approximately 30 °C above the ambient temperature to approximately 100 °C. To stabilize the temperature, a microprocessor-based measurement and control system with an implemented PI controller was designed and implemented. The controller's output signal, in the form of PWM pulses, controlled MOSFET transistors operating in current source mode. The thermal power dissipated in the heat source was calculated in a steady state based on the supply voltage U , the maximum transistor current I_{max} , and the average value of the duty cycle γ , $P = UI_{max}\gamma$. The maximum power dissipated in the heat source was $P_{max} = 40$ W and was supplied to the system in transient state only.

Multiple measurement sessions were conducted for various ambient conditions and heat source power levels. The measurement results from one selected session are presented in Table 1. The tests were conducted in an air preparation chamber for three different ambient parameters (DC_{in}) and different heat source power and temperature levels, as shown in Table 1.

Table 1. Air parameters of the IREC/FFR exchanger with lower water reservoir.

DC_{in} T/RH, °C/%	DC_{out} T/RH, °C/%	HS_{air} T/RH, °C/%	WC_{out} T/RH, /%	P W	T_{HS} °C
28.36/24.81	15.47/55.65	22.21/34.84	23.79/53.45	0.53	29.91
28.45/24.75	15.57/55.40	23.31/33.57	23.84/53.46	2.11	39.62
28.61/24.54	16.19/54.48	24.60/31.65	24.04/53.32	3.65	49.29
28.68/24.52	16.22/54.15	25.37/30.48	24.13/53.35	5.20	58.93
28.74/24.43	16.62/53.69	26.45/29.29	24.22/53.35	6.63	68.49
28.79/24.41	16.86/53.14	27.68/27.69	24.31/53.42	8.30	78.00
28.88/24.31	16.82/52.77	28.78/26.66	24.42/53.42	9.67	87.46
28.94/24.24	17.43/52.15	29.84/25.91	24.54/53.47	11.32	96.83
41.08/16.60	20.95/47.68	30.58/25.58	32.58/45.10	1.47	44.73
41.21/16.55	20.96/47.44	31.49/25.13	32.76/44.95	2.21	49.59
41.30/16.54	20.91/47.12	32.37/24.38	32.87/45.03	3.79	59.23
41.39/16.51	21.62/46.72	33.35/23.50	33.01/45.05	5.36	68.84
41.54/16.48	21.67/46.50	34.65/22.40	33.16/45.03	6.92	78.38
41.80/16.40	22.34/45.78	35.63/21.87	33.43/45.00	8.35	87.87
41.97/16.35	22.35/45.67	36.77/20.89	33.59/44.87	10.03	97.27
54.21/12.42	25.79/43.03	39.28/20.08	41.94/38.03	1.68	54.68
54.35/12.38	25.73/42.70	39.80/19.7	42.09/38.01	2.41	59.52
54.44/12.35	25.79/42.44	40.51/19.2	42.19/38.03	3.98	69.16
54.56/12.35	26.41/42.18	41.57/18.86	42.32/38.10	5.54	78.74
54.56/12.35	26.46/41.90	43.03/17.93	42.37/38.18	7.03	88.26
54.61/12.37	26.60/41.84	43.76/17.72	42.39/38.35	8.59	97.74

Quantitative results of measuring the power dissipated in the MOSFET transistors and the determined thermal resistance values are graphically presented in Figure 6. The measurement results are presented as a function of the temperature difference between the heat source and the temperature at the inlet of the dry channel ($T_{HS} - T_{DCin}$). This way of presenting the results clearly demonstrates the main advantage of using IREC/FFR systems in electronics. The higher the ambient temperature, the better the heat transfer to the environment (better cooling), which is expressed by a lower value of the heat source-to-ambient thermal resistance R_{th} . Furthermore, it is worth noting that for lower values of $T_{HS} - T_{DCin}$, the thermal resistance decreases, ensuring easier energy dissipation to the environment (Figure 6).

An uncertainty analysis of the heat exchanger power and thermal resistance measurement results was conducted. The power P of the heat source was determined using Equation (1). The MOSFET transistors were powered with a voltage of $U = 8$ V, and the average current was regulated by changing the duty cycle γ . The current was determined by measuring the voltage U_s across a shunt with a resistance of $R_s = 10$ m Ω and reduced inductance.

$$P = UI_{max}\gamma = U \frac{U_s}{R_s} \gamma \quad (1)$$

Combined standard uncertainty of power measurement P was determined by Equation (2).

$$u_P = \frac{U}{R_s} \sqrt{(U_s u_\gamma)^2 + (\gamma u_{U_s})^2} \quad (2)$$

where u_γ and u_{U_s} denote the standard uncertainties of the duty cycle γ and the shunt voltage U_s , which are defined by Equation (3).

$$u_\gamma = \frac{s_\gamma}{\sqrt{N_\gamma}}, \quad u_{U_s} = \sqrt{\left(\frac{\Delta U_{smax}}{\sqrt{3}}\right)^2 + \left(\frac{s_{U_s}}{\sqrt{N_{U_s}}}\right)^2} \quad (3)$$

where s refers to the standard deviation and the number of averaged measurements $N_\gamma = 100$ and $N_{U_s} = 16$.

It is worth noting that the duty cycle was determined and averaged directly by the control system, as it was generated using a software PI controller. The shunt voltage was measured using an oscilloscope set to averaging mode.

The thermal resistance R_{th} of the IREC exchanger is defined by Equation (4).

$$R_{th} = \frac{\Delta T}{P} = \frac{T_{HS} - T_{DCin}}{P} \quad (4)$$

where T_{HS} and T_{DCin} are the heat source temperature and the air inlet temperature to the dry channel (DC).

Consequently, the combined standard uncertainty for the thermal resistance depends on the standard uncertainties of the temperature and power measurements (Equation (5)).

$$u_{R_{th}} = \sqrt{\left(\frac{u_{T_{HS}}}{P}\right)^2 + \left(\frac{u_{T_{DCin}}}{P}\right)^2 + \left(\frac{\Delta T}{P^2} u_P\right)^2} = \sqrt{2 \left(\frac{u_T}{P}\right)^2 + \left(\frac{\Delta T}{P^2} u_P\right)^2} \quad (5)$$

It was assumed that the uncertainty of the heat source and air temperature measurement at the dry channel (DC) inlet was the same, as both quantities were measured with the same type of sensor. According to the manufacturer's data, the maximum temperature measurement error was assumed to be $\Delta T = \pm 0.1$ °C, and the standard deviation of $N_T = 100$ measurements was determined at $s_T = 0.05$ °C (Equation (6)).

$$u_T = \sqrt{\left(\frac{\Delta T_{max}}{\sqrt{3}}\right)^2 + \left(\frac{s_T}{\sqrt{N_T}}\right)^2} \quad (6)$$

Based on the considerations presented above, the uncertainties of the power P and thermal resistance R_{th} measurements of the IREC exchanger were determined (Table 2). Finally, the expanded uncertainty of the power P (U_P) and thermal resistance R_{th} ($U_{R_{th}}$) measurements was determined, assuming the value of the coverage factor $k = 1.98$ for the level of confidence $p = 0.95$.

Table 2. Expanded uncertainty of power and thermal resistance of IREC/FFR heat exchanger for electronic application.

T_{DCin} °C	T_{HS} °C	U_P W	R_{th} °C/W	$U_{R_{th}}$ °C/W
28.36	29.91	0.04	3.08	0.13
28.45	39.62	0.06	5.47	0.15
28.61	49.29	0.10	5.8	0.16
28.68	58.93	0.15	6.02	0.17
28.74	68.49	0.21	6.22	0.19
28.79	78.00	0.27	6.17	0.19
28.88	87.46	0.34	6.32	0.22
28.94	96.83	0.43	6.28	0.23
41.08	44.73	0.05	2.66	0.10
41.21	49.59	0.06	3.97	0.12
41.30	59.23	0.11	4.93	0.14
41.39	68.84	0.16	5.34	0.15
41.54	78.38	0.22	5.56	0.17
41.80	87.87	0.29	5.77	0.19
41.97	97.27	0.37	5.79	0.20
54.21	54.68	0.06	0.47	0.05
54.35	59.52	0.07	2.34	0.07
54.44	69.16	0.12	3.90	0.11
54.56	78.74	0.17	4.59	0.14
54.56	88.26	0.24	5.05	0.16
54.61	97.74	0.31	5.27	0.18

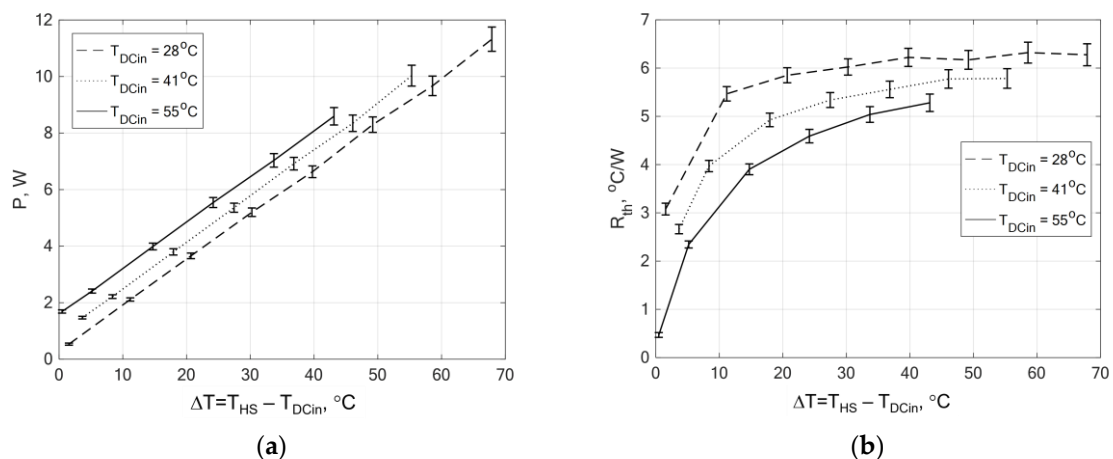


Figure 6. Power dissipated in the heat source: (a) thermal resistance of the electronic system during cooling using the IREC exchanger, as a function of the temperature difference $T_{HS} - T_{DCin}$ (b).

4. Discussion

A more in-depth thermodynamic analysis of the exchanger can be performed using psychrometric diagrams of energy and mass transfer processes (Figure 7). Diagrams are presented for various values of power dissipated in the heat source while maintaining the heat source temperature at approximately 97 °C and for three different ambient temperature values: 28 °C, 41 °C, and 55 °C. The $T_{DCin} - T_{DCout}$ process is air cooling in a dry channel at a constant specific humidity, which determines the mass of water vapor in 1 kg of dry air. The $T_{DCout} - T_{HSair}$ process results from the heating of dry air by the heat source, and the $T_{HSair} - T_{WCout}$ section concerns the change in the values of air parameters in the wet channel—a decrease in temperature and increase in both relative and specific humidity as a result of evaporation.

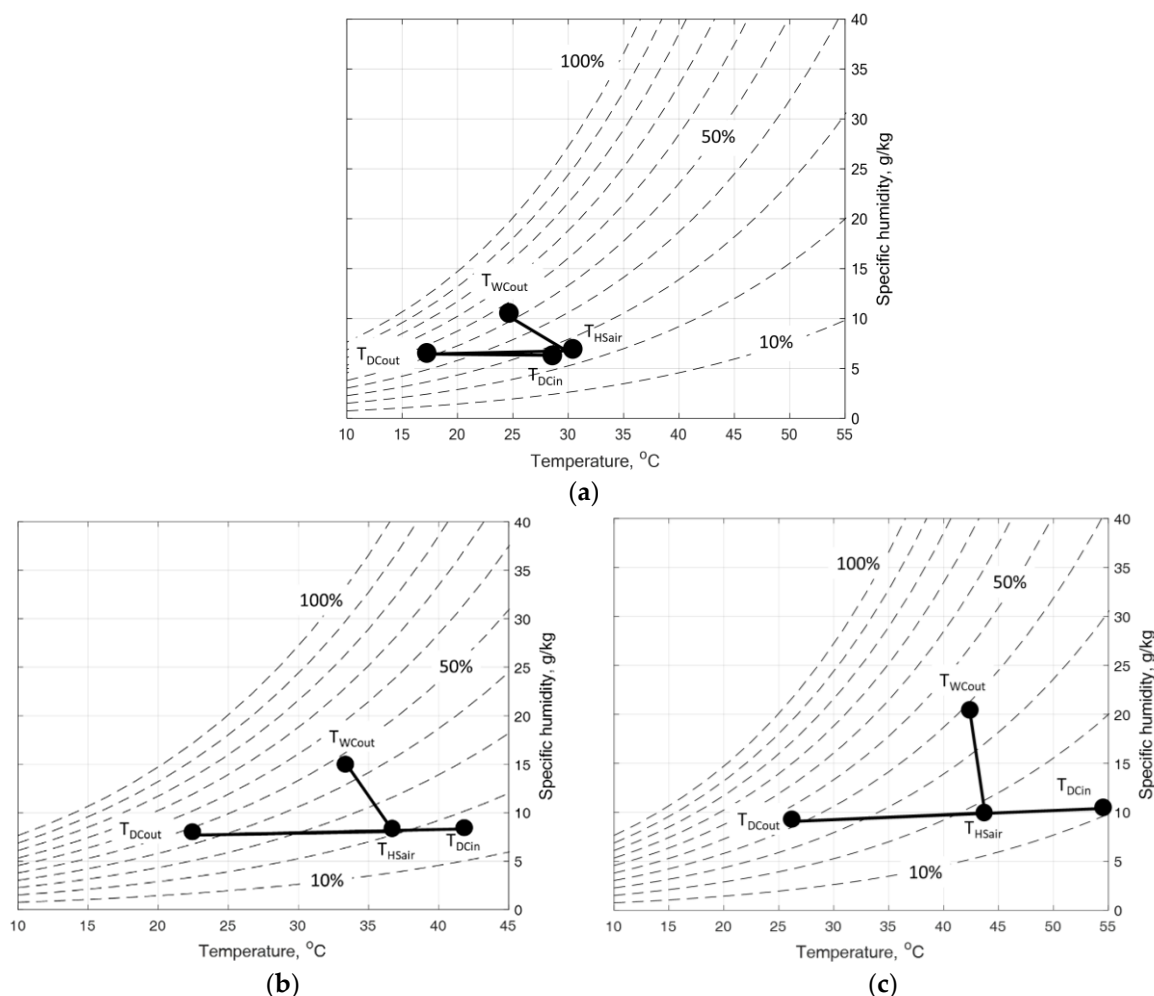


Figure 7. Psychrometric diagrams of thermodynamic processes in IREC/FFR exchangers for ambient temperatures of 28 °C (a), 42 °C (b) and 55 °C (c) for heat source temperature 97 °C.

It is worth noting that in Figure 7, the $T_{DCout} - T_{HSair}$ section, corresponding to air cooling in the dry channel, is not always exactly horizontal. This could lead to the erroneous conclusion that the specific humidity in the dry channel (DC) decreases, which is physically impossible. This result may be due to the incorrect placement of the temperature and humidity sensor, at the upper end of the DC at its outlet, due to the limitations of the exchanger design. In addition, the pressure in the air preparation chamber increases as warm air is pumped in from outside. The additional pressure introduced in the chamber is presented in the psychrometric chart as a slight increase in the specific humidity at the

inlet of the exchanger. This effect can be readily corrected to ensure that all thermodynamic processes presented on the psychrometric chart occur under constant-pressure conditions.

From the psychrometric chart, the increase in specific humidity between the inlet (DC_{in}) and outlet (WC_{out}) air of the exchanger can be estimated. Simultaneously, the change in enthalpy of the air as it is heated by the heat source can be evaluated. All values are normalized to 1 kg of dry air flowing into the exchanger. As a result, the ratio of these two quantities can be used to estimate the water consumption relative to the energy dissipated in the electronics; for example at the inlet temperature of 55 °C and a heat source temperature of 100 °C (Figure 7c) it can reach a level of 2000 J/g.

All sensors were soldered to external frames stiffening the copper membranes coated with porous material in the wet channel (WC). In the next version of the exchanger with a lower water reservoir, the air parameter sensors will be placed in the central part of the channels, which will improve monitoring of the thermodynamic parameters of the air in the exchanger.

Psychrometric diagrams are a simple and effective tool for assessing steady-state thermodynamic phenomena occurring in heat exchangers, where heat, mass, and phase changes occur. They also allow for verification of the design of thermal devices and the presentation of generalized conclusions that will enable their modification and improved efficiency. One of the key parameters for assessing the cooling efficiency of electronic systems using IREC/FFR heat exchangers is the enthalpy change during air heating by the heat source- the $T_{DCout} - T_{HSair}$ process. Table 3 presents the quantitative results (parameter values) of the thermodynamic processes occurring in the heat exchanger operating in the M-cycle.

Table 3. Air parameters of the IREC/FFR-exchanger with a lower water reservoir.

T_{DCin} °C	$T_{DCin} - T_{DCout}$ °C	$H_{DCout} - H_{HSair}$ kJ/kg	P W	T_{HS} °C
54.61	28.01	36.8	8.59	96.83
41.97	20.40	22.9	10.03	97.27
28.94	12.12	11.9	11.32	97.74

The results presented in Table 3 clearly confirm the advantages of using the IREC/FFR cooling system. The largest temperature drop $T_{DCin} - T_{DCout} \approx 28$ °C in the dry channel (DC) was achieved for the highest ambient temperature value $T_{DCin} = 54.61$ °C. At the same time, under the same conditions, the enthalpy change resulting from heat absorption by air and water vapor is the largest and amounts to $H_{DCout} - H_{HSair} = 36.8$ kJ/kg. Consequently, despite the higher ambient temperature value T_{DCin} , the thermal resistance of the exchanger decreased by approx. 20% compared to the thermal resistance of the exchanger for $T_{DCin} = 28.94$ °C ($6.28 \rightarrow 5.27$ °C/W) (Table 3 and Figure 6).

The presented measurement results are consistent with expectations. Compared to forced convection cooling, the IREC/FFR cooling system allows the electronic system to operate at higher power with lower resistance, while maintaining the same heat source temperature. A clear advantage of IREC/FFR cooling can be observed for lower temperature differences between the heat source (T_{HS}) and the environment (T_{DCin}). It is worth recalling that in the absence of a temperature difference between the heat source and the environment, convective cooling disappears completely, and at ambient temperatures higher than the heat source, thermal energy from the environment is transferred to the electronic systems.

Ultimately, the presented experimental results enabled the determination of the IREC/FFR exchanger efficiency indices: η_n , η_a and COP_{IREC} (Equations (7)–(9), Table 4) [27].

Definition of these efficiency factors are given below, but more comments can be found in [27].

Table 4. Performance indicators of the IREC/FFR exchanger with a lower water reservoir at a heat source temperature of $T_{HS} \approx 97^\circ\text{C}$.

$T_{DCin}, ^\circ\text{C}$	$T_{HS}, ^\circ\text{C}$	η_n	η_α	COP_{IREC}
54.61	96.83	−0.11	0.24	4.77
41.97	97.27	−0.41	0.19	5.57
28.94	97.74	−0.65	0.16	6.29
54.21	54.68	0.66	0.36	0.93

The normalized energy efficiency/cooling capacity coefficient of the IREC η_n can vary in the range $<-1,1>$ and is expressed as:

$$\eta_n = \frac{\Delta H_{HS-DCin} - \Delta H_{DCin-DCout}}{|\Delta H_{HS-DCin} + \Delta H_{DCin-DP}|} = \frac{\Delta H_{HS-DCin} - \Delta H_{DCin-DCout}}{|\Delta H_{HS-DP}|} \quad (7)$$

where H refers to enthalpy and DP denotes dew point.

The reduced energy efficiency/cooling capacity ratio compares the cooling capacity of the DC with the cooling potential of the ambient air [27]. Parameter η_n approaches value -1 if no IREC process exists, and $+1$ if only the IREC process dissipates energy from a heat source. It reaches a value of 0 if these two cooling rates are equal. In this case the cooling capacity of the IREC system is twice that of conventional convective cooling due to the additional cooling of the air in the dry channels. It is positive if the IREC gains an advantage. However, a negative value of η_n does not indicate that the IREC system performs worse than conventional cooling under this specific condition. In the IREC system, convective cooling is supplemented by additional heat absorption provided by the cooled air stream flowing through the dry channels.

Furthermore, Table 4 presents the results for a scenario where the heat source and ambient temperatures are nearly equal. Under these conditions, the cooling capacity of the ambient air is negligible. The primary cooling contribution is derived from the IREC process, which is reflected by a high, positive value of η_n . However, COP_{IREC} remains relatively low due to inefficient airflow near the heat source, as further confirmed by the low value of η_α . In summary, while the application of the IREC system is viable in this case, design improvements focusing on optimized airflow and heat removal are required.

The reduced coefficient of heat transfer efficiency from a heat-loaded surface is calculated as follows:

$$\eta_\alpha = \frac{\Delta H_{DCout-HSair}}{\Delta H_{DCout-HS}} \quad (8)$$

A reduced heat transfer efficiency coefficient reflects the efficiency of heat transfer from the heat source to the cold airflow flowing around the electronics. It can be increased by increasing the heat transfer surface, for example, using heat sinks or an additional evaporative layer next to the heat source.

The performance coefficient considering electronics cooling effect is calculated as follows:

$$COP_{IREC} = \frac{P}{P_{fan}} \quad (9)$$

The coefficient of performance allows us to assess how much energy the IREC system can dissipate from the electronic heat source compared to the electrical power supplied to the fan.

5. Conclusions

An indirect evaporative cooling (IREC/FFR) heat exchanger was developed and tested. The IREC technology demonstrator featured an open reservoir in the lower part of the structure and a vertical elevation of water in the wet channels (WCs). The water reservoir was maintained in a fully filled condition during the experiments. The distance between the water surface and the lower edge of the exchanger was less than 5 mm, ensuring continuous wetting of the porous material. Consequently, water was fed upwards to a height of 45 mm in the wet channels (WCs) in an anti-gravity manner. A natural porous material (bamboo fiber) approximately 0.2 mm thick was used, attached with thin double-sided tape to a 0.2 mm thick copper plate separating the wet (WC) and dry (DC) channels.

A simple air preparation chamber with temperature and air supply velocity control was designed and constructed. Holes were made in the upper and lower sections of the chamber to reduce the excess pressure inside. Consequently, tests were conducted at a constant pressure close to atmospheric pressure. The measurements were carried out for various values of the air temperature in the chamber (inlet temperature to the dry channel (T_{DCin})) and various values of the temperature and power dissipated in the heat source (T_{HS} and P), reaching a temperature up to approximately 100 °C and power over 10 W. The experimental results confirmed the operation of the M-cycle heat exchanger and its effectiveness, especially at low values of the temperature difference between the heat source and the environment ($T_{HS} - T_{DCin}$). The values of the heat dissipation efficiency coefficients in electronic systems defined in [27] showed both the potential of the cooling method of heat exchangers operating in the M-cycle and the possibilities of optimizing the exchanger design in order to increase the energy efficiency of the developed solution, which will be the subject of the next research works.

The next step of the research will include improvements to the design of the IREC system, optimized sensor placement, and measurements of pressure drops in the channels, mass flow rate, and other relevant parameters. The system will then be scaled up by increasing the airflow velocity and validated under conditions of higher inlet air relative humidity.

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Abbreviations

The following abbreviations are used in this manuscript:

DP	Dew Point
DC	Dry Channel
WC	Wet Channel
FFR	Full Flow Return
IREC	Indirect Regenerative Evaporative Cooling
HVAC	Heating, Ventilation, and Air Conditioning
PI	Proportional Integral (controller)
PCB	Printed Circuit Board
COP	Coefficient of Performance

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