Pomiary Automatyka Robotyka, ISSN 1427-9126, R. 27, Nr 3/2023, 19–25, DOI: 10.14313/PAR_249/19

Potential of Indirect Regenerative Evaporative Cooling System (M-Cycle) for Electronic Applications

Robert Olbrycht, Marcin Kałuża, Mariusz Felczak, Dmytro Levchenko, Bogusław Więcek
Łódź University of Technology, Institute of Electronics, Wólczańska 211/215, 90-924 Łódź, Poland

Abstract: The article presents a simple prototype system based on the concept of indirect regenerative evaporative cooling (IREC) thermodynamic cycle for electronics applications. The key problem of selecting porous capillary material is discussed and preliminary experimental results are presented using IR thermography. The presented research is an initial step towards the development of a laboratory-validated, fully operational IREC system for high-power electronics.

Keywords: evaporative cooling, porous and capillary materials, heat and mass transfer, IR thermography

1. Introduction

1.1. Maisotsenko thermodynamic cycle

The classical thermodynamic cooling system based on the Maisotsenko cycle consists of two air channels, usually called dry and wet channels (DC and WC) [1–3]. The incoming air is cooled in dry channels (DC) due to intense evaporation process that occurs behind the adjacent wall in wet channels (WC). The evaporating medium (usually water) must be continuously supplied. This is usually done through a thin-film hollow fiber (capillary) membrane separating the two channels. This membrane should conduct heat well to effectively cool dry air. A simplified diagram of the change of the thermodynamic state can be presented using the psychrometric chart (Fig. 1).

The air inlet temperature (A) T_{in} drops due to the heat exchange between the DC and WC and can reach the theoretical dew point limit T_{dp}(B). In practice, it is rather difficult to approach this limit, and the cooling stops earlier (B'). The air in the WC is then wetted to reach saturation (B"). The temperature of moist air and a thin layer of water varies from T_B ($T_{B"}$) to T_C, causing a strong heat flow between the channels and cooling the dry air. To ensure effective cooling, the outlet temperature should be comparable to the inlet temperature. The whole process depends on many parameters, such as mass flow, size of channels, required temperature range, etc.

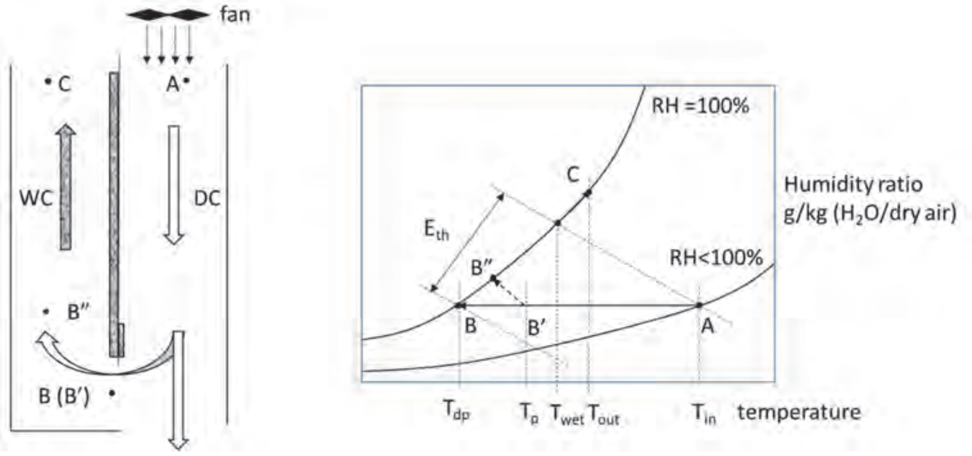


Fig. 1. Typical change of temperature and moisture content during the Maisotsenko cycle
Rys. 1. Typowa zmiana temperatury i wilgotności podczas cyklu Maisotsenko

Let us take an example. Assume that T_{in}= 31 °C and RH 38 %. From the psychrometric chart [33] it is easy to read: T_{dp}= 15.06 °C (dew point), T_{wet} = 20.42 °C (wet bulb). Assuming that the cooling of dry air allows decreasing temperature by 8 °C, thermodynamic parameters vary to RH = 60.9 % and T_{wet} = 17.86 °C. In result the change of enthalpy is about 1.48 kJ/kg. This means that the cooling potential has not been exploited and that there is still room for optimization the configuration of the cooling system.

Autor korespondujący:
Robert Olbrycht, robert.olbrycht@p.lodz.pl

Artykuł recenzowany
nadesłany 11.03.2023 r., przyjęty do druku 24.08.2023 r.

1.2. Applications and porous material membranes

The concept of Indirect Regenerative Evaporative Cooling (IREC), also known as the Maisotsenko or M-cycle [2, 3], was developed and published first in the 1970s. From the beginning, its essence was a heat exchanger with alternating wet and dry channels, with moisture covering the walls of the wet channels, with a part of the airflow at the outlet of the dry channels being redirected to the wet channels. A regenerative heat exchange process was taking place in the device, exploiting the latent heat of water film evaporation in the wet channels due to the psychrometric temperature difference between the cooled dry air and the temperature of a thin layer of water. This temperature difference was the driving force of the heat exchange process in the apparatus, making it possible to cool down the product stream below the wet bulb temperature of the inlet air.

Since then, many research papers, both theoretical and experimental, were published on the subject. So far, IREC solutions were proposed for heating, ventilation and air-conditioning (HVAC) systems, in the power industry for fuel cells, gas turbines, exergy towers or combustion cycles, water distillation/desalination systems or for heat recovery systems [4–10].

Indirect evaporative coolers consume electricity and water to produce cooling energy. Their performance depends on the device geometry and moisture distribution in wet channels. Dew point devices achieve Energy Efficiency Ratio (EER) values of up to 80 %, an order of magnitude greater than typical compressor-based cooling systems [11]. Scientists are striving to reduce the electricity consumption of IREC devices by modifying their geometry [12–14]. Some researchers concentrated on the problem of pressure drop reduction [12, 13] or materials [15]. One of the most important elements of an IREC device is the membrane separating the wet channel from the dry channel. One requires it to be:

- impermeable to fluids, in order to prevent moisture migration from WC to DC; fluid or water vapor migration is allowed only from DC to WC;
- to have low thermal resistance of the porous membrane and high heat transfer coefficient on both sides in order to easily absorb the heat from the DC air and transfer it to the WC, thus accelerating the evaporation and cooling process;
- to provide effective moisture absorption and distribution over its surface from the wet channel side, usually based on capillary effects.

Thus, the use of hydrophobic coatings or porous materials was also investigated in [13, 16, 17]. Whereas not directly devoted to IREC devices and systems, several papers were also lately devoted to the problem of nanoporous membranes wicking and evaporation processes [18–22].

The use of IREC systems for electronics cooling was presented quite rarely and the most informative studies were published in [23, 24, 26]. In [24, 25], the authors investigated the possibility of using IREC technology for data center cooling. In [23], the authors proposed to use IREC technology for electronics cooling, but the content of the paper was general, without electronics-related specifics. In 2021, the article [26] was published. It is the first and so far only paper, where a scaled-down M-cycle cooling system for electronics was presented. It investigated its applicability for electronic device cooling. The ambient air was supplied to the dry channel by a fan. Next, it was cooled down by moisture evaporation in the adjacent wet channel. At the end of the dry channel, a part of the cooled air was drawn by another fan into the wet channel, where it caused evaporation of the water film from the wet sheet separating the channels. The remaining output air from the dry channel was intended to be used for device cooling purposes. The authors demonstrated that the cooler achieved maximum efficiency with turbulent flow in both WC and DC. They also pointed out to the problem of thermal resistance of the wet sheet and the impact of channels pressure drop. This paper sets a very promising direction for further research. However, the proposed solution was used for airflow cooling which next would be used as a cooling medium. Also, the impact of inlet air temperature and humidity was not investigated.

1.3. Tools for modelling phase-change cooling systems

IREC system modeling is widely investigated in the literature. Various types of the IREC models can be found. In general, there can be distinguished analytical and numerical types of the IREC models and both of them can be transient or static. Some of the models consider condensation from the fresh air [27]. Very few papers refer to applications in electronic cooling. Most of the papers assume that the air is perfectly dry. This comes from the fact that IREC systems are widely used for air conditioning in dry regions and the models are related to air conditioning systems. Most of the analytical models are based on simple thermodynamic equations and calculations of the heat and mass transfer through the wet and dry channels. The heat is removed due to the latent heat of water evaporation and thanks to the temperature difference between incoming air and the wall. For numerical simulations various commercial software can be found for example: ANSYS, COMSOL, TRNSYS, etc.

COMSOL is a commercial multipurpose simulation software that can be employed for electromagnetics, structural mechanics, acoustics, fluid flow, heat transfer, chemical engineering and many other applications. The software is based on the FEM (Finite Element Method) which is one of the numerical methods applicable for solving differential equations in many fields of physics. In FEM, an object is divided into small elements. Each element is represented by element equation in order to approximate equations for whole geometry. These small elements are called finite elements. The aim of the method is to minimize an error function and find a stable solution. Evaporative cooling can be simulated in COMSOL environment using Heat Transfer Module. In gases the saturation pressure P_{sat} can be defined. At this pressure, the gas and liquid phase of the substance are in equilibrium. This state depends on the temperature. COMSOL uses the following approximation for P_{sat} (1) [30].

$$P_{sat}\left(T\right) = 610.7 \cdot 10^{7.5\frac{T-273.15}{T-35.85}} Pa \qquad (1)$$

where T is substance temperature.

TRNSYS is commercial software that enables to simulate of transient systems like: central plant modeling, building simulation (including LEED – energy modeling – modeling according to Leadership in Energy and Environmental Design organization regulations), solar thermal processes, ground coupled heat transfer, high-temperature solar applications, geothermal heat pump systems, coupled multizone thermal/airflow modeling, optimization, energy system research, emerging technology assessment, power plants (biomass, cogeneration), hydrogen fuel cell systems, wind and photovoltaic systems, data and simulation calibration. The software has a lot of ready-to-use models like pumps, ducts, and HVAC equipment. The user can build his own model or models can be edited by the user. Evaporation and coming from this evaporation heat transfer can be simulated in TRNSYS software but a precise description of the model could not be found [31].

ANSYS software is also based on FEM concept. The area of applications is really wide. For example, it can be employed for the simulation of: electric motors, electrified powertrain system integration, electromagnetic interface and compatibility, electronics reliability, gas turbines, gas-liquid systems, heat exchangers and many others.

Evaporation and condensation modeling can be done using Volume of Fluid (VOF) modeling [32]. It bases on the assumption that the simulated fluids are immiscible. For example, the momentum equations are solved separately for each of the fluids in the domain. Apart from the modeling of the air as VOF the evaporation-condensation model can be applied. The Eulerian multiphase model is used. The mass transfer for $T_{sat} > T$ takes form of equation (2).

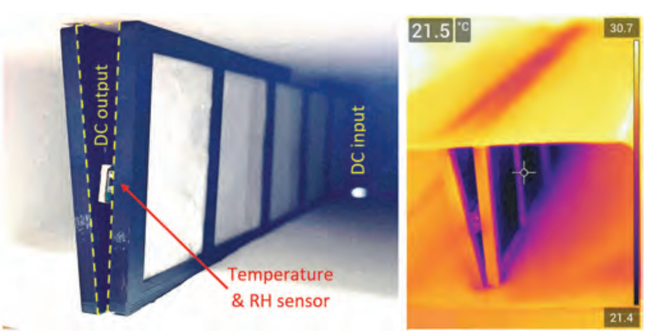
$$m_{e\to v} = coeff \cdot \alpha_v \rho_v \frac{T - T_{sat}}{T_{sat}} \tag{2}$$

where: $coeff$ – coefficient that can be interpreted as a relaxation time [32], α_v and ρ_v – phase volume fraction and density, $m_{e\to v}$ – mass flow rate, T, T_{sat} – temperature and saturation temperature.

The first of the above equations refers to mass transfer from liquid to vapor phase and is expressed in kg/s/m^3. The mass flow rate multiplied by the latent heat gives the source term for the energy equation. ANSYS package also uses the Hertz-Knudsen equation [29] that enables the calculation of evaporation-condensation flux expressed in kg/s/m^2.

$$F = \beta \sqrt{\frac{M}{2\pi R T_{sat}}\left(P^* - P_{sat}\right)} \tag{3}$$

where: P_{sat} – saturation pressure, T_{sat} – saturation temperature, R – universal gas constant, β – accommodation coefficient $1 \geq \beta \geq 0$ (depends on evaporation rate) [32], P^* – vapor partial pressure on the gas side of the interface, M – molar mass.



a) b)

Fig. 2. a) Porous membrane structure inside the prototype, with temperature and relative humidity sensor visible, b) thermal image showing the cooling effect, with 21.5 °C measured in the center point

Rys. 2. a) Porowata struktura membrany wewnątrz prototypu z widocznym czujnikiem temperatury i wilgotności względnej, b) obraz termiczny przedstawiający efekt chłodzenia, przy temperaturze 21,5 °C mierzonej w punkcie środkowym

Analytical models are usually simple and based on heat and mass balance equations well known from thermodynamics. One of the most important issue is to model properly the wet channel. There has to be considered heat and mass transfer between air and the water film. The following formula (4) presenting the gradient of absolute humidity along the wet channel can be found in the literature [28]:

$$\frac{d\omega_{W,f}}{dx} = \frac{h_m A_W}{L_W m_{W,f}}\left(\omega_{W,S} - \omega_{W,f}\right) \tag{4}$$

where: $\omega_{w,s}$ – saturated absolute humidity (at working air temperature $T_{w,f}$), A_W – cross sectional area, L_w – channel length, $m_{w,f}$ – mass flow rate (VOF), h_m – convection coefficient.

2. Prototype cooling system and preliminary experimental results

The prototype was built according to the principle described in section 1.1. However, two wet channels were placed at both sides of the dry channel, instead of only one shown in Fig. 1. It was done to eliminate the need for thermal insulation at one side and to further improve cooling performance. The air flow through the prototype was forced by two low-speed 5 cm-diameter fans. The length of the prototype was 29 cm, with two 27 cm long vertical membranes forming a dry channel, 6 cm high, with vertically variable width, ranging from zero to 1 cm. This variable width enables further investigation of optimal channel width. The membranes, supported by 3D-printed frames, were made of thin metallic layer covered with thin, porous material, as shown in Fig. 2a. The metallic layers were directed to the inside of the dry duct, and the porous ones to the wet one.

Thanks to the capillary effect, the whole membranes area was supplied with water from a tank placed below the bottom plate, creating cooling effect as shown in thermal image taken with a bolometer thermal imaging camera (Fig. 2b). Note the cooling effect is also visible at the top plate, along the DC.

Two temperature and humidity sensors were placed in the prototype – in the beginning and end of the DC. The second one is partially visible in Fig. 2. The sensing elements were not in touch with the structure and measured air parameters – namely temperature and RH, as shown in figures 3 and 5.

There are three input temperature values set during the experiment – 31 °C, 40 °C and 26.5 °C (corresponding to phase 1, 2 and 3). It can be observed that increasing input temperature results in higher difference between input and output temperature, as shown in Fig. 4. It translates to better wet bulb effi-

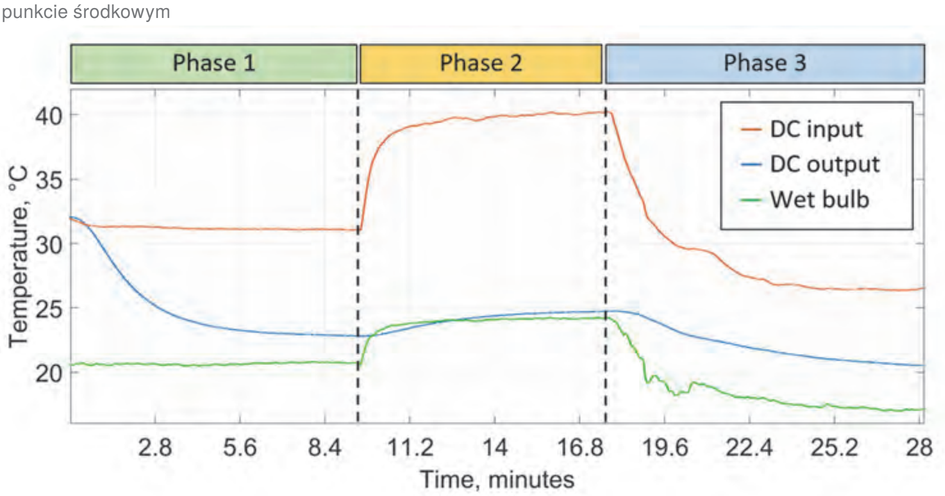


Fig. 3. Temperature chart – input and output of the DC

Rys. 3. Wykres zmian temperatury w czasie na wlocie i wylocie suchego kanału (DC)

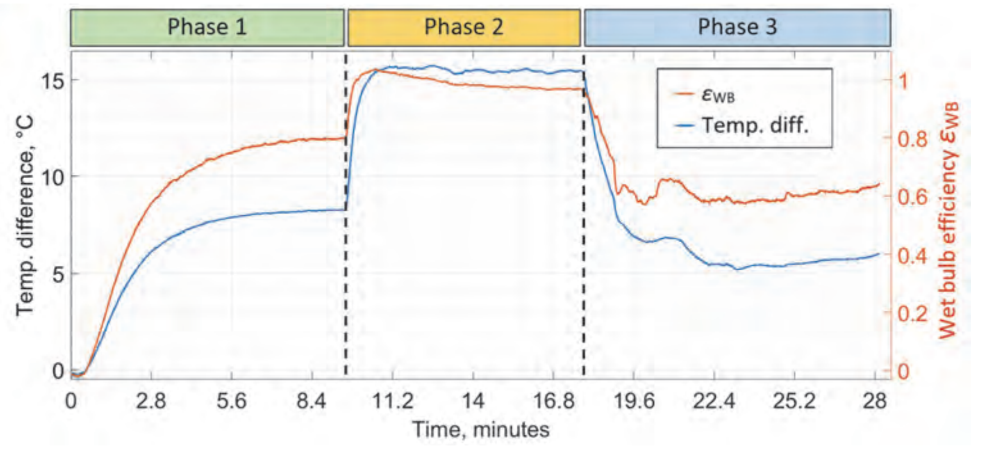


Fig. 4. Chart of temperature difference between input and output of the DC and wet bulb efficiency
Rys. 4. Wykres różnicy temperatury na wlocie i wylocie suchego kanału oraz sprawność procesu odniesiona do temperatury mokrego termometru

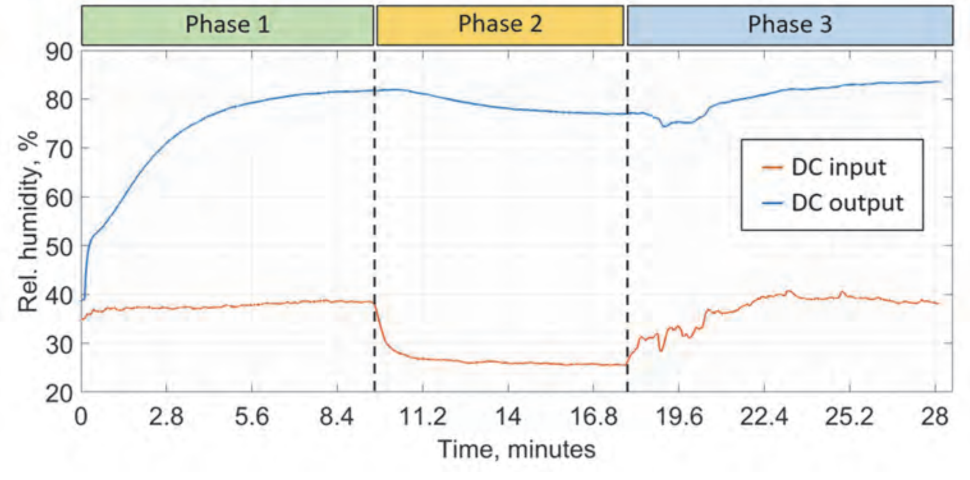


Fig. 5. RH chart – input and output of the DC
Rys. 5. Wykres zmian wilgotności względnej na wlocie i wylocie suchego kanału

ciency ε_{WB} expressed by the formula (5) and shown in Fig. 4. Temperature indexes are used according to Fig. 1.

$$\varepsilon_{WB} = \frac{T_{in} - T_p}{T_{in} - T_{wet}} \quad (5)$$

In terms of relative humidity, it increases along the DC due to cooling. When recalculated to absolute humidity, its change turns out to be negligibly increased at the DC output compared to its input, as there is no water access to DC (except some possible insignificant leaks from WCs).

One can notice WB efficiency, calculated with formula (5), overpassing 1 during phase 2 in Fig. 4. It reaches 1.0378 and results from WB temperature (calculated basing on DC input temperature and RH) exceeding DC output temperature in this phase, as shown in Fig. 3. This effect confirms operation in M-cycle (which is characterized by $\varepsilon_{wb} > 1$, that is calculated with reference to T_{WB} for input parameters) and is correct from the theoretical point of view, as the limit is $\varepsilon_{dp} = 1$.

3. Discussion and conclusions

The data obtained allow us to conclude that the use of IREC technology for cooling electronics has significant potential. However, it should be noted that the efficiency of the heat exchanger, and its geometry in the framework of the experimental campaign was far from optimal, because the main goal of this work is to determine and verify the potential of IREC technology for electronics cooling tasks, as well as to outline the scope of the expected tasks when using this technology.

Figure 2 shows the experimental cell of the heat exchanger in which the cross-section of the dry channel varies in height. This made it possible to evaluate the influence and relationships of the geometry of the dry channel (primarily its height and length) with the achieved temperature of the cooling flow. The wet channel here has been deliberately made significantly larger to minimize the parallel effect of wet channel geometry on cooler performance. The significantly larger wet duct volume in a given test cell maximizes the driving force of the evaporation process, which in a limited wet duct cell volume would invariably drop due to the rapid saturation of the air with evaporated moisture vapor. In addition, based on the data obtained using a thermal imaging camera, it was possible to estimate the temperature gradient along the channel, see Fig. 2b (upper part of the apparatus), which made it possible to confirm the reduction in the driving force of the heat transfer process.

Based on the theory of heat transfer, it is known that the theoretical limiting efficiency of any heat exchanger for a given temperature difference is achieved if its heat transfer surface tends to infinity. Of course, such a condition has no practical application, but based on it, it is possible to determine the optimal geometry of the heat exchanger with a sufficient degree of accuracy, by iterative approximations. Moreover, taking into account that IREC technology devices use water as a refrigerant, which has high values of specific heat and phase transition heat during evaporation, the optimal geometric dimensions of the device should be significantly smaller than devices in which there are no coolant phase transitions. The only “weak link” in apparatuses of this type is a dry channel, where phase transitions do not occur, the heat capacity and thermal conductivity of the air are low, and the gas flow rates are not high. Therefore, it is very important to determine the practical limit of the length of the heat exchanger using the IREC technology, based primarily on the geometry of the dry channel (its width). To do this, it is rational to step by step change the length of the device at a constant value of the width of dry and wet channels at each iterative approximation. For example, it is possible to evaluate the characteristics of a cooler taking the width of a dry channel in the range 1–3 mm with an iteration step of 0.5 or 1 mm and device lengths of 250, 500, 750, 1000, 1250 mm, respectively, for each iteration. If, at the selected geometric ratio (dry channel width), the temperature reached at the outlet of the cooler ceases to decrease with an increase in the length of the apparatus, it can be stated that the practical limit of the heat exchanger according to the IREC technology has been reached using the selected materials and the selected layout.

Based on the experimental data presented in Figures 3 and 4, it can be concluded that it is preferable to conduct tests for environmental conditions with increased temperature, when the need for cooling electronic devices increases. The recommended minimum temperature level at the inlet to the apparatus can be taken as 32 °C, because starting from this temperature, the need for cooling electronic devices increases significantly. It is important to note that ambient humidity plays a key role in determining the effectiveness of IREC devices, as it is the ratio of partial pressures of air at the inlet to the wet channel that is the key driving force of the evaporation process. Therefore, it is advisable to clarify the optimal geometry of the apparatus, taking into account the influence of the humidity of the incoming air on its characteristics. The recommended minimum humidity level is proposed to be 40 %. This level of humidity corresponds to most of the real conditions for cooling electronic devices in a room (indoor conditions). As can be seen from Fig. 5, the output relative humidity of the air does not exceed 90 %, which eliminates the phenomenon of condensation on the surface of electronic devices and is an additional positive feature of heat exchangers using IREC technology.

In conclusion, it should be noted that to date there are no publications in which experimental studies have been carried out to determine the limits of the cooling capacity of the IREC technology for electronics. Therefore, further study of this technology can provide important knowledge both about the fundamental processes and determine the practical limits of its applicability for cooling electronics. As part of the next experimental campaign, it is planned to create a more advanced experimental stand for studying the characteristics of IREC heat exchangers with different geometries.

Acknowledgments

The work was supported by the National Science Centre, Poland, under research project „Phase change electronic devices cooling systems based on the thermodynamic Maisotsenko cycle„number 2022/45/B/ST7/02820.

References

1. Levchenko D., Yurko I., Artyukhov A., Baga V., *Maisotsenko cycle applications for multistage compressors cooling.* IOP Conference Series: Material Science and Engineerings, Vol. 233, 2017, DOI: 10.1088/1757-899X/233/1/012023.
2. Maisotsenko V. et. al., *Method of indirect-evaporative air cooling in household electrical appliances*, Proceedings of World Electrotechnical Congress, Moscow, USSR, 1977.
3. Maisotsenko V. et. al., *Air Cooling Device for Indirect Evaporative Cooling*, USSR Patent 979,796.
4. Mahmood M.H., Sultana M., Miyazaki T., Koyama S., Maisotsenko V.S., *Overview of the Maisotsenko cycle – A way towards dew point evaporative cooling*, "Renewable and Sustainable Energy Reviews", Vol. 66, 2016, 537–555, DOI: 10.1016/j.rser.2016.08.022.
5. Dizaji H.S., Hu E.J., Chen L., *A comprehensive review of the Maisotsenko-cycle based air conditioning systems*, "Energy", Vol. 156, 2018, 725–749, DOI: 10.1016/j.energy.2018.05.086.
6. Sajjad U. et al., *A review of recent advances in indirect evaporative cooling technology*, "International Communications in Heat and Mass Transfer", Vol. 122, 2021, DOI: 10.1016/j.icheatmasstransfer.2021.105140.
7. Gorshkov V., *Systems Based on Maisotsenko Cycle: Coolerado Cooler*, BSc thesis, Building Services Engineering, Mikkeli University of Applied Sciences, December 2012.
8. Pacak A., Worek W., *Review of Dew Point Evaporative Cooling Technology for Air Conditioning Applications*, "Applied Sciences", Vol. 11, No. 3, 2021, DOI: 10.3390/app11030934.
9. Bi Y., Wang Y., Ma X., Zhao X., *Investigation on the Energy Saving Potential of Using a Novel DewPoint Cooling System in Data Centres*, "Energies", Vol. 10, No. 11, 2017, DOI: 10.3390/en10111732.
10. Pandelidis D., Cichoń A., Pacak A., Anisimov S., Drąg P., *Performance comparison between counter and cross-flow indirect evaporative coolers for heat recovery in air conditioning systems in the presence of condensation in the product air channels*, "International Journal of Heat and Mass Transfer", Vol. 130, 2019, 757–777, DOI: 10.1016/j.ijheatmasstransfer.2018.10.134.
11. Duan Z. et. al., *Indirect evaporative cooling: Past, present and future potentials*, "Renewable and Sustainable Energy Reviews", Vol. 16, No. 9, 2012, 6823–6850, DOI: 10.1016/j.rser.2012.07.007.
12. De Antonellis S., Cignatta L., Facchini C., Liberati P., *Effect of heat exchanger plates geometry on performance of an indirect evaporative cooling system*, "Applied Thermal Engineering", Vol. 173, 2020, DOI: 10.1016/j.applthermaleng.2020.115200.
13. Xu P., Ma X., Zhao X., Fancey K., *Experimental investigation of a super performance dew point aircooler*, "Applied Energy", Vol. 203, 2017, 761–777, DOI: 10.1016/j.apenergy.2017.06.095.
14. Waqas M.A. et. al., *Performance enhancement of a cross flow dew point indirect evaporative cooler with circular finned channel geometry*, "Journal of Building Engineering", Vol. 35, 2021, DOI: 10.1016/j.jobe.2020.101980.
15. Jia L., Liu J., Wang C., Cao X., Zhang Z., *Study of the thermal performance of a novel dew point evaporative cooler*, "Applied Thermal Engineering", Vol. 160, 2019, DOI: 10.1016/j.applthermaleng.2019.114069.
16. Lee J., Lee D.-Y., *Experimental study of a counter flow regenerative evaporative cooler with finned channels*, "International Journal of Heat and Mass Transfer", Vol. 65, 2013, 173–179, DOI: 10.1016/j.ijheatmasstransfer.2013.05.069.
17. Lv J., Xu H., Xu T., Liu H., Qin J., *Study on the performance of a unit dew-point evaporative cooler with fibrous membrane and its application in typical regions*, "Case Studies in Thermal Engineering", Vol. 24, 2021, DOI: 10.1016/j.csite.2021.100881.
18. Ranjan D. et al, *Vapor generation via porous nanochannel wicks*, "Cell Reports Physical Science", Vol. 3, No. 2, 2022, DOI: 10.1016/j.xcrp.2022.100738.
19. Liu R., Liu Z., *Enhanced Evaporation of Ultrathin Water Films on Silicon-Terminated Si_3N_4 Nanopore Membranes*, "Langmuir", Vol. 37, No. 3, 2021, 10046−10051, DOI: 10.1021/acs.langmuir.1c0121.
20. Xia G., Wang J., Zhou W., Ma D., Wang J., *Orientation effects on liquid-vapor phase change heat transfer on*

nanoporous membranes, "International Communications in Heat and Mass Transfer", Vol. 119, 2020, DOI: 10.1016/j.icheatmasstransfer.2020.104934.
21. Hanks D.F., Lu Z., Sircar J., Salamon T.R., Antao D.S., Bagnall K.R., Barabadi B., Wang E.N., *Nanoporous membrane device for ultra high heat flux thermal management*, "Microsystems & Nanoengineering", Vol. 4, No. 1, 2018, DOI: 10.1038/s41378-018-0004-7.
22. Li R., Wang J., Xia G., *New Model for Liquid Evaporation and Vapor Transport in Nanopores Covering the Entire Knudsen Regime and Arbitrary Pore Length*, "Langmuir", Vol. 37, No. 6, 2021, 2227−2235, DOI: 10.1021/acs.langmuir.1c00023.
23. Maisotsenko V., Reyzin I., *The Maisotsenko Cycle for Electronics Cooling*, 2005 International Electronic Packaging Technical Conference and Exhibition, IPACK2005, 415–424, DOI: 10.1115/IPACK2005-73283.
24. Weerts B.A. et. al., *Green Data Center Cooling: Achieving 90% Reduction: Airside Economization and Unique Indirect Evaporative Cooling*, 2012 IEEE Green Technology Conference, USA, DOI: 10.1109/GREEN.2012.6200950.
25. Bi Y., Wang Y., Ma X., Zhao X., *Investigation on the Energy Saving Potential of Using a Novel Dew Point Cooling System in Data Centres*, "Energies", Vol. 10, No. 11, 2017, DOI: 10.3390/en10111732.
26. Dizaji H.S. et. al., *Proposing the concept of mini Maisotsenko cycle cooler for electronic cooling purposes; experimental study*, "Case Studies in Thermal Engineering", Vol. 27, 2021, DOI: 10.1016/j.csite.2021.101325.
27. Chen Y., Luo Y., Yang H., *A simplified analytical model for indirect evaporative cooling considering condensation from fresh air: Development and application*, "Energy and Buildings", Vol. 108, 2015, 387–400, DOI: 10.1016/j.enbuild.2015.09.054.
28. Tariq R., Benarab F.Z., *Mathematical Modelling and Numerical Simulation of Maisotsenko Cycle*, World Academy of Science, Engineering and Technology, "International Journal of Mechanical and Mechatronics Engineering", Vol. 11, No. 9, 2017, 1618–1625.
29. Persad A., Ward Ch., *Expressions for the Evaporation and Condensation Coefficients in the Hertz-Knudsen Relation*. "Chemical Reviews". Vol. 116, No. 14, 2016, 7727-8314, DOI: 10.1021/acs.chemrev.5b00511.

Other sources

30. www.comsol.com – Introduction to Modeling Evaporative Cooling
31. www.trnsys.com/
32. www.ansys.com – Resource center, Ansys 2021 R1: Multiphase Volume of Fluid (VOF), Evaporation-condensation Model
33. www.kwangu.com/work/psychrometric.htm

Możliwości zastosowania systemu IREC (pracującego w cyklu M) do chłodzenia układów elektronicznych

Streszczenie: W artykule przedstawiono prototypowy układ chłodzenia oparty na koncepcji cyklu termodynamicznego pośredniego regeneracyjnego chłodzenia wyparnego (IREC) do zastosowań w elektronice. Omówiono kluczowy problem doboru porowatego materiału kapilarnego i przedstawiono wstępne wyniki eksperymentów z wykorzystaniem termografii w podczerwieni. Przedstawione badania stanowią wstępny krok w kierunku opracowania zweryfikowanego laboratoryjnie, w pełni funkcjonalnego systemu IREC do odprowadzania ciepła w systemach elektronicznych dużej mocy.

Słowa kluczowe: chłodzenie wyparne, materiały porowate i kapilarne, wymiana ciepła i masy, termografia w podczerwieni

Robert Olbrycht, PhD, Eng.

robert.olbrycht@p.lodz.pl
ORCID: 0000-0001-9842-2815

Robert Olbrycht received the Ph.D. degree in electronic engineering from the Lodz University of Technology, Lodz, Poland, in 2012. He is currently an Assistant Professor of Electronics with the Institute of Electronics, Lodz University of Technology. He has published papers on optical gas imaging with thermal imaging cameras and applications of thermography for industrial purposes.

Marcin Kałuża, PhD, Eng.

marcin.kaluza@p.lodz.pl
ORCID: 0000-0003-2990-1702

Marcin Kałuża was born in Lodz, Poland. He received the M.Sc. and Ph.D. degrees in electronics from Lodz University of Technology in 1995 and 2007, respectively. He is an Assistant Professor at Lodz University of Technology. His research interests include thermal management of electronic circuits, temperature measurements, and industrial applications of thermography.

Mariusz Felczak, PhD, Eng.

mariusz.felczak@p.lodz.pl
ORCID: 0000-0003-2360-6091

Mariusz Felczak received the MSc degree in electronic engineering from Lodz University of Technology, Poland, in 2002. Earlier he carried out research for his master thesis on thermal measurements in electronics at the ELIS Department of Ghent University, Belgium. Since 2002 he is working in Lodz University of Technology. He received the PhD degree from the Lodz University of Technology in Poland in 2007. His research interests include modelling of heat and mass exchange in electronics cooling.

Dmytro Levchenko, PhD, Eng.

dmytro.levchenko@p.lodz.pl
ORCID: 0000-0003-0766-4371

Received the Ph.D. degree in mechanical engineering from the Sumy State University, Sumy, Ukraine, in 2012. He currently performs as an Assistant Professor of Ecotechnology Team, Lodz University of Technology, Poland. His activities cover the research and development of modern energy efficient cooling, heating and recovery systems, as well as the improvement and development of evaporative cooling technologies for various technical applications.

Prof. Bogusław Więcek, DSc, PhD, Eng.

boguslaw.wiecek@p.lodz.pl
ORCID: 0000-0002-5003-1687

Prof. Bogusław Więcek is the head of the Department of Electronic Circuits and Thermography at the Institute of Electronics of the Lodz University of Technology. He specializes in computer thermography and modeling of thermal problems in electronic systems. He is a member of the scientific committee of the international Conference on Quantitative Infrared Thermography (QIRT) and chairman of the Conference on Infrared Thermography and Thermometry organized periodically in Poland.