

Analysis of Heat Transfer Coefficients for a Micro-Jet Heat Exchanger

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Abstract – This study investigated a microjet heat exchanger. The microjet heat transfer technique offers numerous advantages, such as high heat transfer coefficients, efficient heat transfer even at low temperature differences, the ability to easily regulate the heat flux, and the possibility of direct contact with the cooled surface. The experimental setup and the microjet heat exchanger are discussed. The experiments were conducted using a microjet-generating head equipped with four membranes, each having 364 circular slits with a diameter of 160 μm , a nozzle-to-heat-transfer-surface area ratio A_r of 0.0019, and a ratio of the distance between adjacent nozzles to their diameter s/d of 19. The working fluid was water. The volumetric flow rate was adjusted manually and ranged from 12.8 to 133 mL/s, corresponding to Reynolds numbers in the range of 279–4166. The experiments were conducted for four different distances between the nozzles and the heat transfer surface, adjusted using appropriate spacers with thicknesses of 1, 3, 5, and 7 mm. The heat transfer coefficients k_{exp} were calculated from the obtained results.

Keywords – Energy efficiency; heat transfer coefficient; microjets; microjet heat exchanger.

Nomenclature		
d	Nozzle diameter	m
H	Distance between the nozzles and the surface	m
k	Total heat transfer coefficient	W/(m ² K)
Re	Reynolds number	–
S	Nozzle's pattern	m
T	Temperature	°C
V	Volumetric flow rates	mL/s
Subscripts		
c	cold	
exp	experimental	
h	hot	

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1. INTRODUCTION

Heat transfer using microjets can be divided into two types: free-surface jets and submerged jets. In both cases, the medium is forced through nozzles to form jets [1]. Free-surface jets involve a fluid flowing into a gaseous environment [2]. In the type using submerged jets, the medium is a fluid that flows into the same fluid in the same state of matter. This distinction is important due to the different dependence of heat transfer on variable parameters for these two types of jets, as well as differences in how the jets interact with each other. As reported by the authors of [3], [4], increasing the flow rate and the distance between the nozzles and the cooled surface does not significantly affect heat transfer in the case of free jets, unlike submerged jets [5]. Fig. 1 illustrates the division into regions characteristic of microjets, using a single free jet as an example. The first of these is the stagnation region, formed as a result of the jet's impact and deflection, and the second is the wall jet region. The highest heat transfer coefficients are achieved in the vicinity of the stagnation region; however, the wall jet region can also have a significant impact on heat transfer. This will be particularly evident in the case of interactions between adjacent jets, resulting in the appearance of a local maximum of the Nusselt number between them. The terminology used also applies to submerged jets; however, it is worth noting that such a jet, due to the much stronger effects of the surrounding fluid, takes on a different shape, as can be seen in Fig. 2.

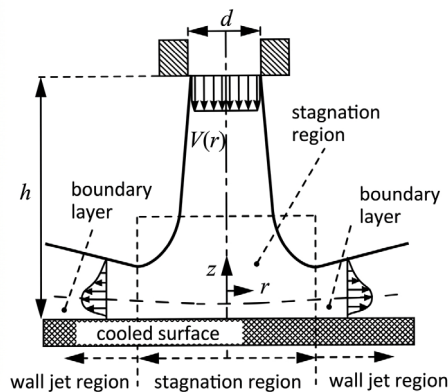


Fig. 1. Characteristic regions for a free-flowing stream [6].

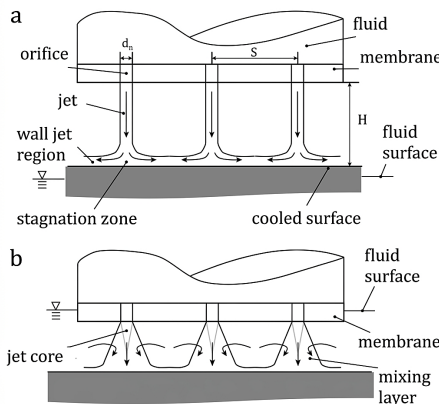


Fig. 2. Types of streams: a) free-flowing streams; b) submerged streams [2].

A separate issue concerns single-stream heat exchangers, which will not be analyzed in detail in this paper. Single submerged streams in the stagnation zone can achieve higher local Nusselt numbers and heat transfer coefficients than those obtained with stream grids; however, to practically solve the problem of maintaining acceptable surface temperatures and temperature gradients, the latter option must be used [7]. Typically, the distribution of the heat transfer coefficient for a single stream follows a bell-shaped curve, with a local maximum in the stagnation zone. The coefficient decreases radially with increasing distance from the stagnation point; however, researchers often observe a renewed increase in the coefficient in the region at a distance of 1.5–2.5 nozzle diameters.

Based on studies describing the behaviour of multiple microjets [6], [8], [9], it can be observed that the distribution of the heat transfer coefficient in this case is much more complex and, in particular, is sensitive to changes in the distance between adjacent nozzles and the distance between the nozzles and the cooled surface [10], [11]. These relationships have been investigated experimentally and numerically in numerous studies [1], [5], [12]–[14].

When cooling using a jet array, interactions occur between adjacent jets. The first type of disturbance is cross-flow, generated by the fluid ejected from the upper nozzles when the heat transfer surface is oriented vertically, or by the nozzles located in the centre of the array; flowing toward the downstream nozzles, this flow disrupts their operation and also mixes with subsequent jets. This effect can be observed in numerical simulations conducted by Alabbagh and Mohammad in [6]. The authors analysed the effect of varying the distance between the nozzles and the heat transfer surface (H/B) for five linearly arranged square nozzles using air as the medium at a Reynolds number of 200. The study was conducted for the following parameters: $2 \leq H/B \leq 20$; $S/B = 4$; $R = \{0.5; 1; 2.5; 5; 7.5; 10\}$, where B is the nozzle side length, S is the distance between adjacent nozzles, and R is the ratio of jet velocity to cross-flow velocity.

Another type of flow and heat transfer disturbance is the mutual interaction of jets on the heat transfer surface. This results in the appearance of a local maximum of the Nusselt number between two adjacent nozzles. This phenomenon was described by the authors of [4], who investigated the influence of the H/d and s/d parameters for air using a head generating five microjets. Four nozzles were arranged at the corners of a square, and the fifth in its centre.

The jets also interact with each other before reaching the surface onto which they strike [15]. This occurs for small values of the s/d parameter and simultaneously high values of the H/d parameter. The first effect of this is the entrainment effect, which causes the jets to deflect toward the centre of the system. This is caused by the central jet drawing in the surrounding fluid. The second phenomenon is the mixing of the jets, which results in the Nusselt number distribution becoming more uniform while simultaneously reducing its average value. The authors of the paper [4], [14] state that as the s/d parameter increases, and as the H/d parameter increases, the interaction between adjacent jets decreases sharply. Increasing distance H results in thorough mixing and the aforementioned uniform distribution of the heat transfer coefficient. At the same time, increasing the distance between the nozzles makes the areas cooled by the corresponding jets distinct, though these are disrupted by the influence of cross-flow, which reduces the Nusselt number for the stagnation zones of the outer jets.

2. PRESENTATION OF THE RESEARCH STATION

Two streams of water – hot and cold, both drawn from the mains – were fed to the shower head. The flow rate of both fluids was adjusted manually using two valves located on the

pipes supplying water to the shower head. Temperature measurement was performed using four thermocouples located at the inlet and outlet of the hot and cold water. The results were recorded at a frequency of 50 Hz and displayed on a computer screen as a graph. Based on this, the duration of a single measurement was determined, after which the mass flow rate was adjusted. After obtaining a sufficient number of measurements, the head was disassembled, new spacers were inserted, and the measurements started over from the beginning. The microjet-generating head can be used as a co- or counter-current heat exchanger. Its universal design allows for easy membrane replacement, enabling adjustment of the number, shape, size, and arrangement of the holes (optimization of the parameters d , s/d , Ar). Furthermore, the system of interchangeable spacer plates allows for adjusting the distance between the nozzles and the heat transfer surface (H/d). The head consists of three chambers (fluid at the same temperature is supplied to the two outer segments), which allows the heat transfer surface area to be doubled compared to a two-chamber heat exchanger. The experiment utilized a microjet-generating head, which served as a co-current exchanger, and the circuit diagram shown in Fig. 3.

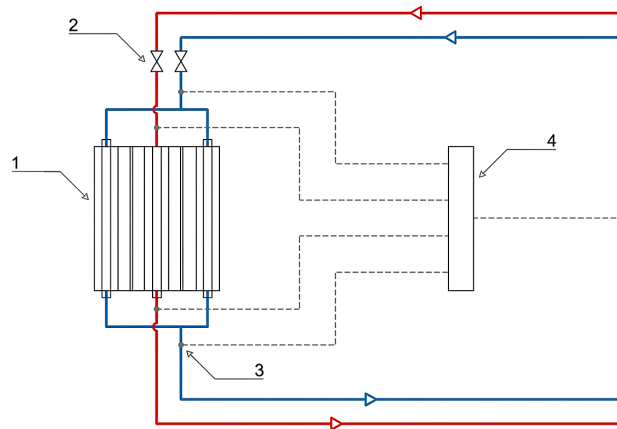


Fig. 3. Test setup diagram: 1 – head; 2 – valve; 3 – thermocouple; 4 – data acquisition card.

The operating principle of the heat exchanger involves supplying two fluid streams at different temperatures. One stream is directed to the central chamber, while the other splits and flows simultaneously into the two outer chambers. The fluid is forced through four diaphragms at the same time, forming micro-streams that strike a 1-mm-thick aluminium partition. The heat exchange surface is a 63×63 mm square. The distance between the membrane and the partition is adjusted by installing appropriate spacers. As the fluid flows down the partition, it absorbs or releases heat and is discharged to the outside.

This study investigated a co-current heat exchanger using water as the medium. A membrane with 364 holes, each 160 μm in diameter, was used; the holes were arranged in a linear pattern at equal intervals in both directions ($\Delta x = \Delta y = 3 \text{ mm}$). Experiments were conducted for spacer thicknesses of 1, 3, 5, and 7 mm. The volumetric flow rates of cold and hot water were manually adjusted within the range of $12.8 \leq \dot{V} \leq 133.3 \text{ [mL/s]}$, which approximately corresponds to Reynolds numbers of $145 \leq Re \leq 2275$.

Fig. 4 shows a diagram of the membrane, one of the three heat exchanger segments, and photos of the head. In Fig. 5, you can see the outer and middle segments separated by two

5-mm-thick spacers and a 1-mm-thick aluminium plate. The photo also shows the membrane bolted to the middle segment of the head.

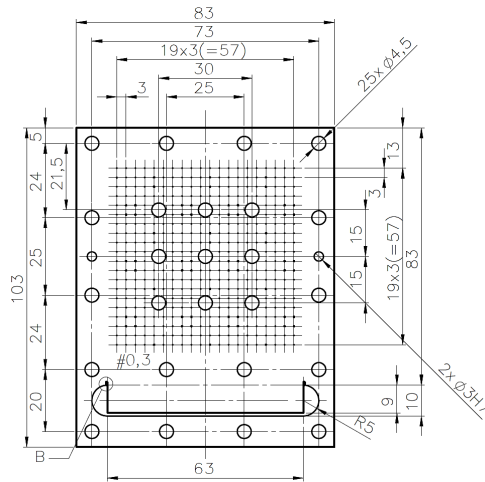


Fig. 4. Membrane generating microjets.

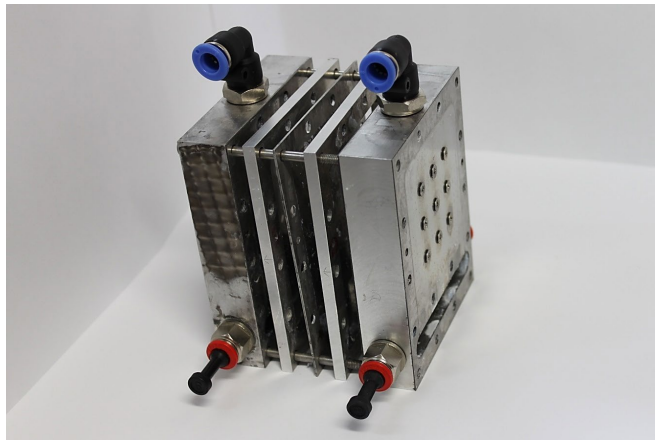


Fig. 5. Components between the outer and middle segments.

3. RESULTS OF THE EXPERIMENTAL STUDIES

The measurement results are presented in the form of Tables 1–4 and Fig. 6–9 included later in this chapter. The tables show the measured volumetric flow rates for the hot and cold sides of the heat exchanger, the inlet and outlet temperatures on the hot and cold sides of the heat exchanger, as well as the calculated heat transfer coefficient k_{exp} . The graphs were formatted so that the main vertical axis displays the experimentally determined heat transfer coefficient k_{exp} , while the auxiliary axis on the right shows the Reynolds number, Re . The Reynolds number is plotted separately for the hot side, Re_h , and the cold side, Re_c . Analysis of the results reveals a positive effect of increasing the Reynolds number on the heat transfer coefficient.

TABLE 1. MEASUREMENT RESULTS FOR H = 1 MM

Lp.	$\dot{V}_h$, mL/s	$\dot{V}_c$, mL/s	$T_{h,inlet}$, °C	$T_{h,outlet}$, °C	$T_{c,inlet}$, °C	$T_{c,outlet}$, °C	k_{exp} , W/(m ² K)
1	34.86	16.10	49.46	45.03	11.45	28.08	4235.98
2	65.26	15.57	48.77	45.90	10.55	28.58	4733.20
3	84.21	15.33	49.93	47.30	9.70	29.43	5114.02
4	114.06	15.62	48.87	46.94	10.10	30.31	5448.76
5	28.03	42.38	52.21	36.99	8.27	18.36	7721.28
6	54.55	42.73	50.44	42.09	8.07	21.30	8939.10
7	74.13	42.21	50.02	43.13	7.76	22.40	9980.22
8	122.77	42.80	51.13	46.04	8.02	25.23	11 782.44
9	70.44	86.11	50.39	40.57	7.25	16.52	12 011.07
10	84.57	83.78	50.61	41.71	6.68	17.06	12 994.93
11	120.72	83.46	50.90	44.12	6.94	18.80	14 340.66
12	36.21	101.40	51.39	35.15	6.68	13.07	10 254.73
13	68.85	100.76	50.76	39.71	6.67	15.19	12 955.84
14	123.81	101.92	50.85	43.28	6.70	17.44	15 767.29

TABLE 2. MEASUREMENT RESULTS FOR H = 3 MM

Lp.	$\dot{V}_h$, mL/s	$\dot{V}_c$, mL/s	$T_{h,inlet}$, °C	$T_{h,outlet}$, °C	$T_{c,inlet}$, °C	$T_{c,outlet}$, °C	k_{exp} , W/(m ² K)
1	16.03	12.84	52.53	45.66	11.29	19.39	1766.65
2	32.49	12.79	48.66	45.66	11.10	20.63	1947.85
3	53.45	13.04	49.60	46.99	10.59	20.65	2298.44
4	40.32	37.27	50.51	40.24	8.15	20.18	7917.14
5	62.50	37.50	49.65	42.56	8.41	21.68	8610.65
6	96.72	36.89	51.38	45.47	7.99	24.30	10 364.76
7	126.56	36.70	51.06	46.48	8.19	25.37	10 777.98
8	43.71	71.50	52.06	39.33	6.99	15.77	9777.19
9	58.44	71.31	49.97	40.30	7.66	16.39	10 087.37
10	92.49	70.69	50.80	43.18	7.51	18.50	12 240.21
11	41.41	100.68	51.17	37.00	7.23	13.16	9861.18
12	75.88	103.75	51.07	40.94	6.93	14.94	12 775.02
13	122.92	101.71	50.96	43.64	7.57	16.81	14 612.36

TABLE 3. MEASUREMENT RESULTS FOR H = 5 MM

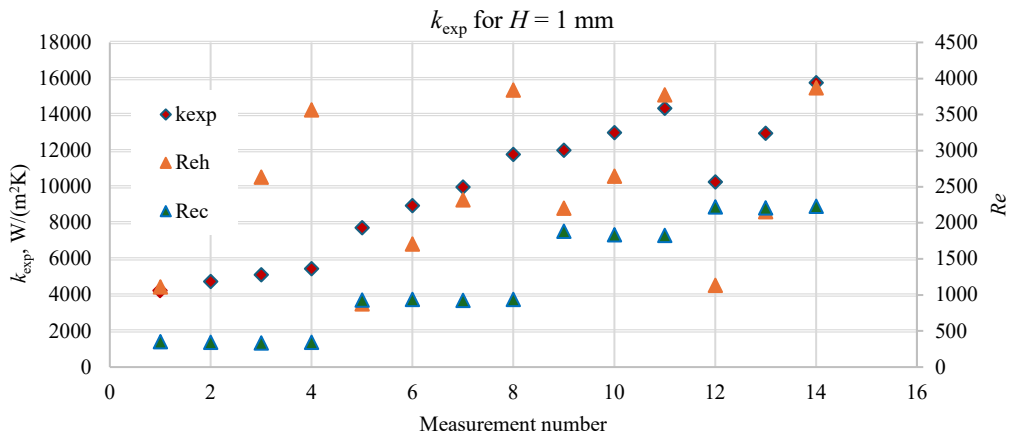
Lp.	$\dot{V}_h$, mL/s	$\dot{V}_c$, mL/s	$T_{h,inlet}$, °C	$T_{h,outlet}$, °C	$T_{c,inlet}$, °C	$T_{c,outlet}$, °C	k_{exp} , W/(m ² K)
1	10.92	17.91	50.26	41.33	9.71	14.63	1829.32
2	20.00	18.52	51.26	44.52	9.28	16.54	2051.12
3	45.36	17.75	49.26	44.13	9.40	21.29	4038.63
4	58.21	17.88	47.87	44.39	9.38	21.70	3924.17
5	72.58	17.90	48.40	45.29	9.43	23.31	4558.39

6	91.89	17.53	50.50	47.71	10.15	26.09	5037.22
7	35.10	31.88	51.12	40.19	8.33	20.88	7411.92
8	51.03	31.17	50.45	42.69	8.39	22.74	8076.08
9	73.21	31.96	50.00	44.73	8.59	23.95	8112.52
10	112.00	32.11	50.06	46.73	8.51	26.08	8611.36
11	42.54	94.37	50.40	37.79	7.17	13.46	9806.41
12	66.86	100.81	50.72	41.63	7.42	14.46	10 301.12
13	130.43	95.36	50.29	44.43	7.34	16.39	13 106.19

TABLE 4. MEASUREMENT RESULTS FOR $H = 7$ MM

Lp.	$\dot{V}_h$, mL/s	$\dot{V}_c$, mL/s	$T_{h,inlet}$, °C	$T_{h,outlet}$, °C	$T_{c,inlet}$, °C	$T_{c,outlet}$, °C	k_{exp} , W/(m ² K)
1	30.77	13.66	50.34	46.47	12.43	21.35	2123.05
2	37.80	41.88	51.28	39.67	8.57	19.18	7995.00
3	69.30	40.95	51.24	44.49	8.04	21.37	8640.47
4	52.77	65.83	51.05	43.61	7.62	16.61	7793.88
5	67.55	65.74	50.51	42.74	7.79	16.81	9103.48
6	108.84	66.67	52.28	46.10	7.10	18.47	10 979.96
7	122.56	65.74	52.17	46.79	7.39	19.45	11 251.77
8	35.19	100.00	51.27	36.24	7.35	12.85	9073.10
9	56.81	102.76	51.27	40.40	7.21	13.99	10 440.36
10	80.26	98.34	51.78	43.01	6.84	14.31	10 869.99
11	133.33	90.42	51.18	44.59	6.74	15.56	12 685.08

For a distance of $H = 1$ mm, Fig. 6 was prepared based on experimental data. Heat transfer coefficients ranging from 4235 W/(m²K) to 15 767 W/(m²K) were obtained; higher values corresponded to higher water flow rates through the heat exchanger and, consequently, to higher Reynolds numbers, which ranged from 875 to 3868 for hot water and 385–2229 for cold water.

Fig. 6. Experimental results for the heat transfer coefficient k_{exp} for $H = 1$ mm.

For a distance $H = 3$ mm between the microjets and the heat transfer surface, Fig. 7 was prepared based on experimental data. Heat transfer coefficients ranging from 1766.7 W/(m²K) to 14 612.4 W/(m²K) were obtained; with higher values corresponding to higher water flow rates through the heat exchanger and, consequently, higher Reynolds numbers, which range from 524 to 3840 for hot water and from 279 to 2269 for cold water.

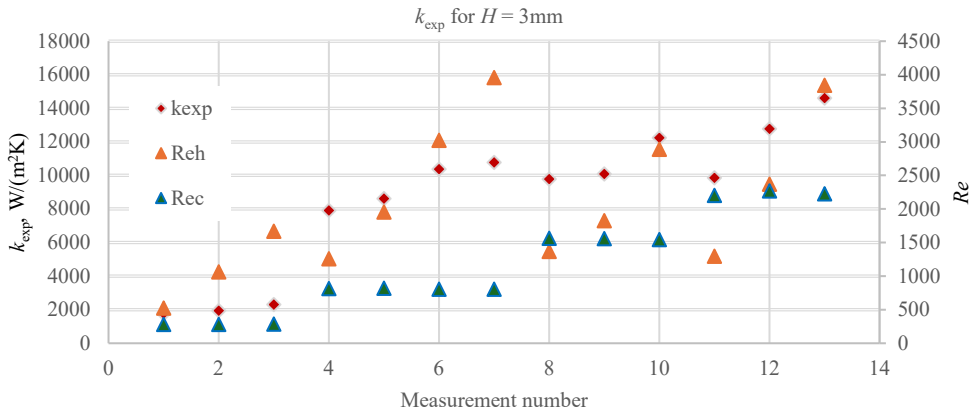


Fig. 7. Experimental results for the heat transfer coefficient k_{exp} for $H=3$ mm.

For a distance $H = 5$ mm between the microjets and the heat transfer surface, Fig. 8 was prepared based on experimental data. Heat transfer coefficients ranging from 1829.3 W/(m²K) to 13 106.2 W/(m²K) were obtained; higher values corresponded to higher water flow rates through the heat exchanger and, consequently, to higher Reynolds numbers, which ranged from 357 to 4075 for hot water and from 383 to 2205 for cold water.

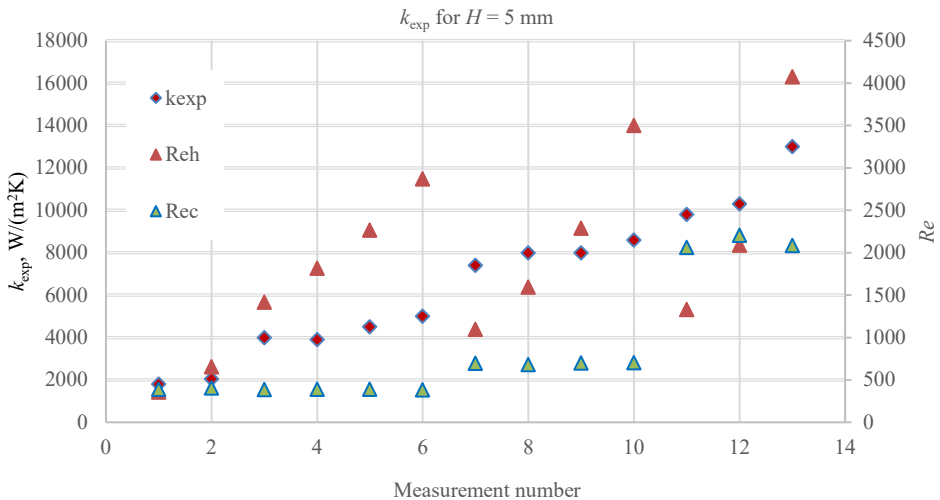


Fig. 8. Experimental results for the heat transfer coefficient k_{exp} for $H = 5$ mm.

For a distance of $H = 7$ mm between the microjet and the heat transfer surface, heat transfer coefficients in the range of 2123.1–12685.1 W/(m²K) were determined experimentally. The Reynolds numbers for hot water ranged from 1006 to 4166, and for cold water from 298 to 2247.

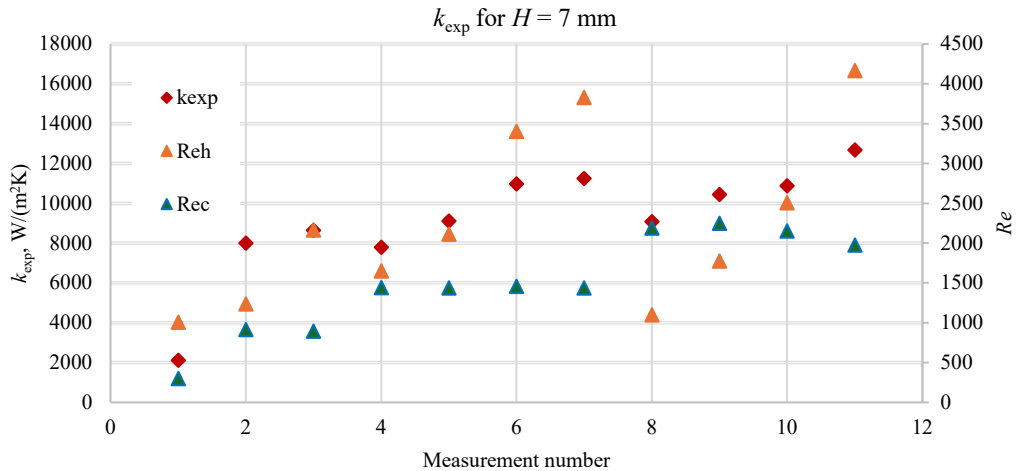


Fig. 9. Experimental results for the heat transfer coefficient k_{exp} for $H = 1$ mm.

In the case of heat transfer coefficients, it can be observed that the value of this coefficient decreases as the distance between the microjets and the heat transfer surface increases. The Reynolds number depends primarily on the fluid flow, and the characteristic dimension – the diameter of the microjet – remains constant during the experiments; furthermore, the fluid's viscosity values do not change significantly within the given temperature range.

4. CONCLUSION

Experimental studies yielded heat transfer coefficients ranging from $1533.6 \leq k_{exp} \leq 15\,767.3$ W/(m²K) at an average temperature difference between the hot and cold water inlets of 42.3 °C. The volumetric flow rate was varied in the range from 12.8 to 133.3 mL/s, corresponding to Reynolds numbers from 279 to 4166. Using spacers of varying thickness, measurements were taken for four different distances between the membrane and the heat exchange surface – 1, 3, 5, and 7 mm. An increase in the heat transfer coefficient was observed alongside a simultaneous increase in mass flow rate on both sides of the exchanger. It was also noted that even large changes in flow rate, in the case of low flow on one side of the exchanger, do not have a significant effect on the k_{exp} value. Since the water flow was controlled manually, only for a few measurements was it possible to achieve volumetric flow rates that would allow for a comparison of k_{exp} values for different values of the H parameter. For measurements where a comparison is possible, an increase in the heat transfer coefficient is observed as the distance from the heat exchange surface decreases, with this relationship being more pronounced for higher volumetric flow rates.

The microjet heat exchanger under investigation provides intensive and easily adjustable heat transfer [16]–[22], and its well-thought-out design allows for easy modification of individual geometric parameters. The heat exchanger should be the subject of further research

to determine its optimal operating parameters, although even in its current form it competes effectively with other devices of this type.

Microjet cooling technology, with its numerous advantages – including, above all, high heat transfer coefficients, the maintenance of a uniform temperature distribution across the heat exchange surface, and the possibility of regulation – is the subject of extensive research and is constantly evolving. This method responds to the growing demand for compact and efficient heat removal methods.

REFERENCES

- [1] Mikieliewicz D., Mikieliewicz J. Surface cooling using axially symmetric liquid jets. Gdańsk, Wydawnictwo Politechniki Gdańskiej, 2005. (In Polish)
- [2] Grzebielec A., Rusowicz A., Szelągowski A. Air purification in industrial plants producing automotive rubber components in terms of energy efficiency. *Open Engineering* 2017;7:1. <https://doi.org/10.1515/eng-2017-0015>
- [3] Robinson A. J., Schnitzler E. An experimental investigation of free and submerged miniature liquid jet array impingement heat transfer. *Experimental Thermal and Fluid Science* 2007;32(1):1–13. <https://doi.org/10.1016/j.expthermflusci.2006.12.006>
- [4] San J.-Y., Chen J.-J. Effects of jet-to-jet spacing and jet height on heat transfer characteristics of an impinging jet array. *International Journal of Heat and Mass Transfer* 2014;71:8–17. <https://doi.org/10.1016/j.ijheatmasstransfer.2013.11.079>
- [5] Łapka P., Ciepliński A., Rusowicz A. Numerical model and analysis of heat transfer during microjets array impingement. *Energy* 2020;203. <https://doi.org/10.1016/j.energy.2020.117879>
- [6] Aldabbagh L. B. Y., Mohamad A. A. Effect of jet-to-plate spacing in laminar array jets impinging. *Heat and Mass Transfer* 2007;43:265–273. <https://doi.org/10.1007/s00231-006-0109-7>
- [7] Angioletti M., Di Tommaso R. M., Nino E., Ruocco G. Simultaneous visualization of flow field and evaluation of local heat transfer by transitional impinging jets. *International Journal of Heat and Mass Transfer* 2003;46(10):1703–1713. [https://doi.org/10.1016/S0017-9310\(02\)00479-9](https://doi.org/10.1016/S0017-9310(02)00479-9)
- [8] Browne E. A., Michna G. J., Jensen M. K., Peles Y. Experimental investigation of single-phase microjet array heat transfer. *Journal of Heat Transfer* 2010;132(4). <https://doi.org/10.1115/1.4000888>
- [9] San J.-Y., Lai M.-D. Optimum jet-to-jet spacing of heat transfer for staggered arrays of impinging air jets. *International Journal of Heat and Mass Transfer* 2001;44(21):3997–4007. [https://doi.org/10.1016/S0017-9310\(01\)00043-6](https://doi.org/10.1016/S0017-9310(01)00043-6)
- [10] Muszyński T., Andrzejczyk R. Applicability of arrays of microjet heat transfer correlations to design compact heat exchangers. *Applied Thermal Engineering* 2016;100:105–113. <https://doi.org/10.1016/j.applthermaleng.2016.01.120>
- [11] Kotrys-Działak D. Efficiency of Heat Exchanger Systems Working with Nanofluids. *Environmental and Climate Technologies* 2026;30(1):122–141. <https://doi.org/10.2478/rtuct-2026-0009>
- [12] Michna G. J., Browne E. A., Peles Y., Jensen M. K. The effect of area ratio on microjet array heat transfer. *International Journal of Heat and Mass Transfer* 2011;54(9–10):1782–1790. <https://doi.org/10.1016/j.ijheatmasstransfer.2010.12.038>
- [13] Fan Ye, Jian Sun, Jie Li, Renping Zhang. Comparative Study on Flow and Heat Transfer Characteristics of Four Array Rib Micro-Jet Heat Sinks Based on Chip Cooling. *ASME Journal of Thermal Science and Engineering Applications* 2026;18:101011. <https://doi.org/10.1115/1.4071342>
- [14] Garimella S. V., Nenaydykh B. Nozzle-geometry effects in liquid jet impingement heat transfer. *International Journal of Heat and Mass Transfer* 1996;39(14):2915–2923. [https://doi.org/10.1016/0017-9310\(95\)00382-7](https://doi.org/10.1016/0017-9310(95)00382-7)
- [15] Pavlenko A. Study of the Influence of Temperature and Pressure on the Intensity of Gas Hydrate Formation. *Environmental and Climate Technologies* 2024;28(1):724–737. <https://doi.org/10.2478/rtuct-2024-0056>
- [16] Li C.-Y., Garimella S. V. Prandtl-number effects and generalized correlations for confined and submerged jet impingement. *International Journal of Heat and Mass Transfer* 2001;44(18):3471–3480. [https://doi.org/10.1016/S0017-9310\(01\)00003-5](https://doi.org/10.1016/S0017-9310(01)00003-5)
- [17] McGlen R. J., Jachuck R., Lin S. Integrated thermal management techniques for high power electronic devices. *Applied Thermal Engineering* 2004;24:1143–1156. <https://doi.org/10.1016/j.applthermaleng.2003.12.029>
- [18] Meola C. A new correlation of Nusselt number for impinging jets. *Heat Transfer Engineering* 2009;30(3):221–228. <https://doi.org/10.1080/01457630802304311>
- [19] Tie P., Li Q., Xuan Y. Investigation on the submerged liquid jet arrays impingement cooling. *Applied Thermal Engineering* 2011;31:2757–2763. <https://doi.org/10.1016/j.applthermaleng.2011.04.048>
- [20] Womac D. J., Incropera F. P., Ramadhyani S. Correlating equations for impingement cooling of small heat sources with multiple circular liquid jets. *Journal of Heat Transfer* 1994;116:482–486. <https://doi.org/10.1115/1.2911423>

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- [21] Whelan B. P., Robinson A. J. Nozzle geometry effects in liquid jet array impingement. *Applied Thermal Engineering* 2009;29:2211–2221. <https://doi.org/10.1016/j.applthermaleng.2008.11.003>
- [22] Yuan-Yang L, Zhen-Hua L, Guo-Shan W., Le P. Experimental study on critical heat flux of high-velocity circular jet impingement boiling on the nano-characteristic stagnation zone. *International Journal of Heat and Mass Transfer* 2013;67:560–568. <https://doi.org/10.1016/j.ijheatmasstransfer.2013.08.058>