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A MICROJET BASED RECUPERATOR FOR APPLICATION IN DOMESTIC MICRO CHP

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ABSTRACT

In the paper the original design of a compact heat exchanger with microjets producing intensification effect is presented. Its primary application is for the domestic Organic Rankine Cycle (ORC), however, the design is universal and may have numerous other applications. The technology of microjets manufacturing is an "in-house" patented design. In the present paper the idea of such a heat exchanger is shown together with the flow and thermal characteristics of the prototype. The developed prototype of heat exchanger is capable of exchanging 5 kW of thermal energy at a logarithmic mean temperature difference (LMTD) of 60 K. The total heat transfer surface equal to 0.0072 m^2 leads to very significant heat fluxes. Measured overall heat transfer coefficient reaches 12000 W/m^2K, which was calculated using the Wilson method. The description of the Wilson technique used for the determination of the heat transfer coefficient is also presented in the body of the text. That method seems to be, in the authors' opinion, the only one for finding the heat transfer coefficient for such a complex heat exchanger structure. In this case measurements of wall temperatures are not possible and hence the determination of heat transfer coefficient is difficult. The results of performed measurements are satisfactory and encourage for further research of the original design.

INTRODUCTION

Recently, an increasing importance of so called dispersed generation, based on the local energy sources and technologies utilizing both the fossil fuels and renewable energy resources, is observed worldwide and in the countries of European Union (EU). Generation of electricity on a small domestic scale together with production of heat can be obtained employing gas engine units, micro gas turbines, fuel cells with efficient electrolysis, Stirling engines or ORC systems. Some of the technologies mentioned above are mature and have demonstrated their potential performance and production costs, whereas others are at their earliest stages of market entry and are still relatively expensive and may not perform to their expected potential. The ORC technology is currently produced in small amounts, but is likely to become significantly cheaper over the next few years as it enters mass production. The authors are at the moment involved in a large scale domestic project with the objective of developing a commercially available ORC CHP unit for domestic applications. This technology is also likely to be fully mature much earlier than other promising technologies such as Stirling engines and fuel cells, as the components for micro ORC are commercially available. With the advent of these technologies the (electrical) efficiency levels, currently at around the 10% mark, are likely to rise considerably to reach 25%. In the case of ORC units the power of expanding device of CHP ranges from a few to tens of kilowatts.

NOMENCLATURE

A	[m^2]	surface
h	[J/kgK]	enthalpy
k	[W/m^2K]	heat transfer coefficient
$\dot{m}$	[kg/s]	mass flux
P	[Pa]	pressure
$\dot{Q}$	[W]	rate of heat
T	[K]	temperature
$\dot{V}$	[m^3/s]	volumetric flow
w	[m/s]	velocity

Greek letters

α	[W/m^2K]	convective heat transfer coefficient
δ	[m]	wall thickness
λ	[W/mK]	thermal conductivity

Subscripts

c	cold
h	hot
in	inlet
log	logarithmic
out	outlet

Micro CHP units based on ORC fit very well to the strategy of wider penetration of disperse energy sources and in recent years this technology has become a field of intense research. It appears as a promising technology for conversion of heat into useful work and electricity, because the heat source, in case of disperse systems, can be of various origin, for example solar power, biomass combustion, ground heat source or waste heat. Unlike the steam power cycle, where vapour of water is the working fluid, the ORC employs refrigerants, hydrocarbons, solvents or other organic substances.

Primary energy in such CHP units is better utilized than in units producing only electricity. CHP units utilize the energy of fuel, in almost 85-95%. About 70% of the energy is delivered as heat, and approximately 15 to 25% is additional production of electric energy, as shown in Figure 1. The heat from the power plant can be used for heating of utility hot water for domestic use and central heating, while electricity can be used on site or sold to the grid. In the authors opinion that is by far the best utilization of the chemical energy contained in the fuel [1]. Conventional power plants producing electricity, in some cases, achieve the rate of conversion of over forty percents. In the near future, cycles with efficiency exceeding 50% are expected, but the transmission losses ought to be considered. Therefore the micro CHP units, planned to be developed in the near future, are quite promising technologies. In the literature the several studies of different micro-ORC systems designed for heat production and electricity generation could be found, for example Nguyen et al. [2], Saitoh et al. [3], Lemort et al. [4]. There are still issues requiring particular attention in appropriate implementation of that technology in practice. One of these issues is connected with the expansion device, which in the above cases was a modified scroll compressor to the expansion mode.

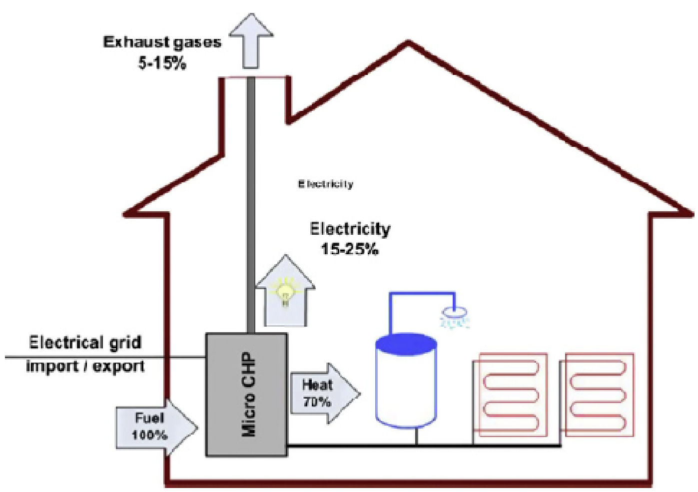

Figure 1 A general scheme of a micro CHP unit

The practical utilization of ORC in micro scale represents a technical challenge. Apart from the completely new design of an expansion machine, the installation should be equipped with highly efficient and small heat exchangers such as evaporator and condenser. In case of so called dry fluids (i.e. when the saturated steam line has a positive slope) then also another regenerator is necessary in the system. The authors acquired knowledge and gained experience in this topic investigating the prototype micro power plant of ORC [1]. In the laboratory they have been studied before commercially available compact heat exchangers and have also developed their own prototype shell-and-tube heat exchangers constructed using the minichannel technology [5].

In this paper, for the purpose of investigating domestic micro power plants and also future technical applications, an original compact heat exchanger with microjets was proposed. The technology of microjets manufacturing is a patented design developed by Plata [6]. The idea of such heat exchanger is shown together with the flow and thermal characteristics of the prototype. Experimental data was also collected and the heat transfer coefficient was calculated using the Wilson method.

MICROJET HEAT EXCHANGER

In this paper the microjet heat exchanger of recuperator type is under consideration. Its consisted of 28 steel membranes with cut microchannels, forming the microjets. Their length was 2.5 mm, width 200 μm and depth 100 μm, respectively, which results in relative small overall pressure drops. The membranes were placed between the plates made of aluminum sheets. The total heat transfer surface was equal to 0.0072 m^2. The membranes and plates had holes, through which the working media could flow and exchange the thermal energy. Each circuit consisted of 1120 microchannels. The structure of described above heat exchanger is shown in Figure 2. The heat exchanger prototype applied in the studies is presented in Figure 3. Figure 4 shows the operating principles of analyzed microjet based heat exchanger.

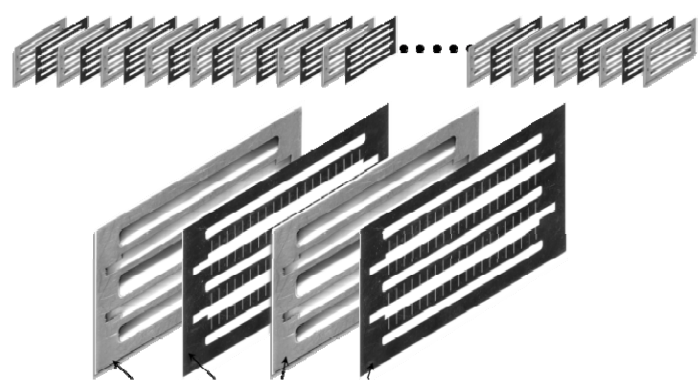

Figure 2 Structure of microjet heat exchanger

The heat transfer process takes place through the membrane which separates two fluids, fixed to body by means of the resin. The membrane is bombarded by microjets impinging from both sides of membranes, producing in this way a highly intensified heat transfer.

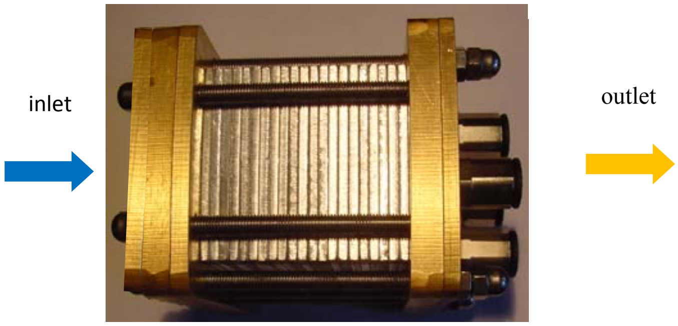

Figure 3 View of prototype microject heat exchanger

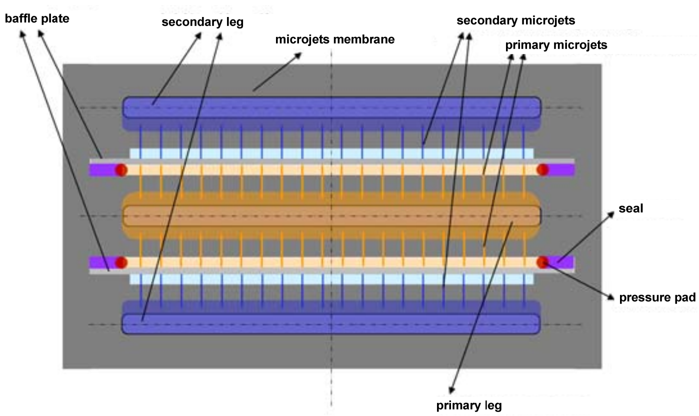

Figure 4 Arrangement of flows in the microjet heat exchanger

EXPERIMENT

The experimental investigations of microjet prototype heat exchanger were carried out on a dedicated facility for testing of heat exchangers, Figure 5.

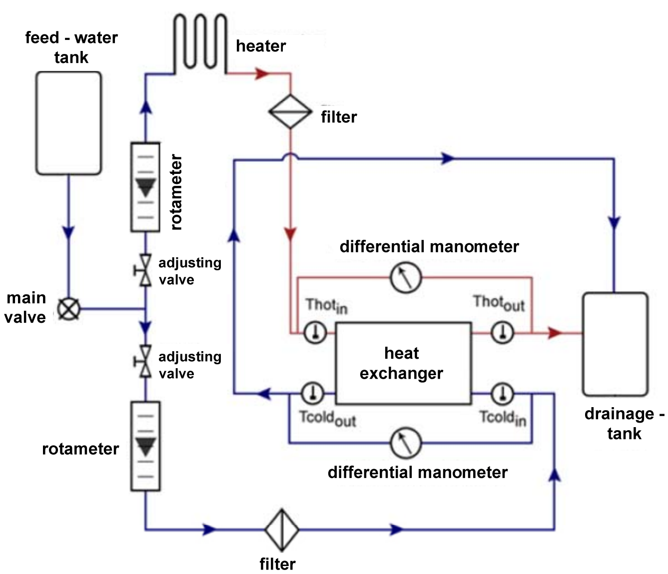

Figure 5 Schematic of experimental facility

The test stand enabled the heat transfer by convection between the hot and cold water. The hot water was circulating in the system with the electric flow heater, while the cold water was a tap water. In both circuits fine filters were installed. The heat was transferred due to the cocurrent flow of working media. The fluid flows were measured by rotameters with accuracy of ±3 l/h. The heater was controlled by the power supply in the range from 0% to 100% of heating power. As a variable parameter the input temperature of heat exchanger was taken and setting up. The pressure drop was measured with mercury manometers with accuracy of ±2 mmHg. Thermocouples of J-type were used to measure temperature in four points i.e. at the inlet and outlet of heat exchanger cold side and at the inlet and outlet of heat exchanger hot side. The reference temperature for thermocouples measurements was equal to 0 ^{o}C and the accuracy of measurements was ±0.5 ^{o}C. Prior to experiments all thermocouples were calibrated to yield the accuracy of measurements of ± 0.5 ^{o}C.

RESULTS OF MEASUREMENTS

During experiments the following parameters were measured: the hot water temperature at the inlet ($T_{hot\ in}$) and at the outlet ($T_{hot\ out}$) of heat exchanger, the cold water temperature at the inlet ($T_{cold\ in}$) and at the outlet ($T_{cold\ out}$) of heat exchanger, the pressure drop connected with the hot water flow (ΔP_h), the pressure drop connected with the cold water flow (ΔP_c), the volumetric flow rate of hot water ($\dot{V}_h$) and the volumetric flow rate of cold water ($\dot{V}_c$). The volumetric flow rate of hot/cold water was varied in the range of 25-250 l/h. The water supply pressure was 4 bar. On the basis of measurement results the rate of heat exchanged between the fluids ($\dot{Q}$), the LMTD in the heat exchanger (ΔT) and the overall heat transfer coefficient (k) were calculated. The heat transfer coefficient was determined with Peclet law based on the rate of heat, taken up by cold water ($\dot{Q}_c$) and the heat transfer area equal to 0.0072 m^2. The characteristics of the rate of heat versus the LMTD is shown in Figure 6. Flow characteristics for hot and cold fluids is presented in Figure 7.

THE WILSON METHOD

As mentioned earlier the experimental study of recuperators require determination of mean heat transfer coefficients on both sides of the wall separating fluids exchanging heat. Usually that requires installation of thermocouples for measurements of wall temperature separating two fluids. If the recuperator has a large number of tubes and a complex surface geometry then accurate measurement of the mean surface temperature faces significant difficulties such as for example in the course of disassembling installation a large number of thermocouples must be attached and subsequently everything must be put up together. Such difficulties can be alleviated if the Wilson's method [7] is applied. The method is very simple and can be applied to the analysis of different types of heat exchangers [8]. A simple and efficient Wilson method in a version similar to the original one is presented below. The classical Wilson method, as well as its modifications, requires only determination of the overall thermal resistance in the heat exchanger. From it an accurate energy balance, based on measurement of flow rates of fluids exchanging heat and their mean temperatures at inlet and outlet of the heat exchanger.

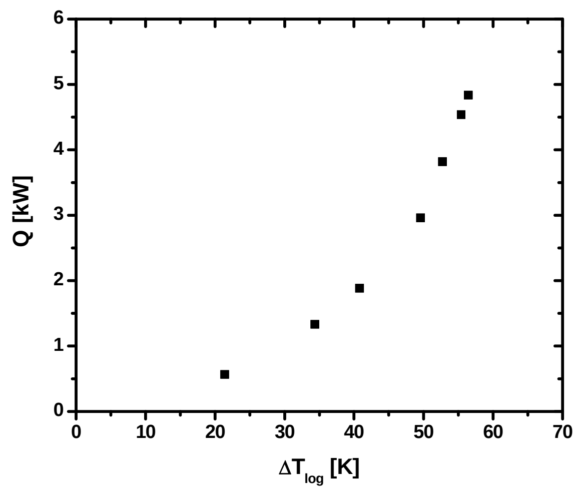

Figure 6 Rate of heat versus log-mean temperature

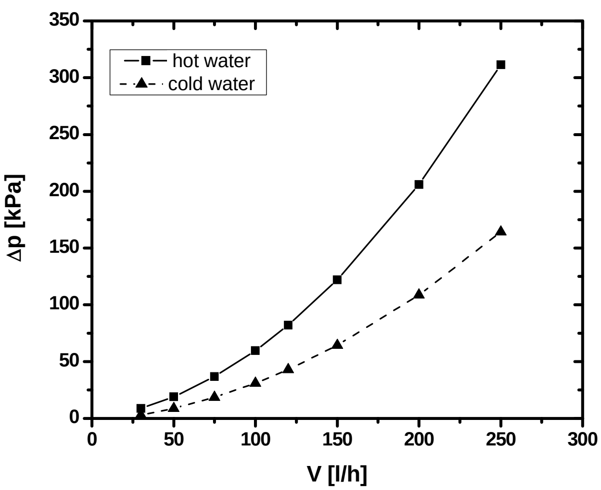

Figure 7 Flow characteristics for hot and cold circuit

Let's consider the simplest case of heat transfer in a recuperator and its corresponding temperature profile (Figure 8). In the recuperator exchanging heat between the single phase fluids (possible are different cases of flows of liquid and gas) the following plan of experiments is assumed. In the first series of experiments the flow rate of one fluid (the one of interest) is kept constant, whereas there are measurements at varying flowrate of the other fluid. On the other hand, in the second series of tests the opposite assumption about the flowrates is required, i.e the flow rate of the second fluid is sustained constant and the flow rate of first fluid varies.

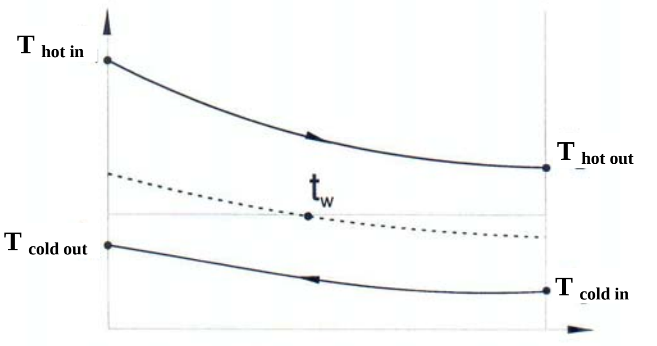

Figure 8 Counter-current temperature distribution in heat exchanger

Assuming that the overall heat transfer coefficient k is known and determined from the thermal balance in the form:

$$\dot{Q} = k\Delta T_{\log} A = \dot{m}_1 \left(h_1 - h_2\right) = \dot{m}_2 \left(h_4 - h_3\right) \tag{1}$$

where: $\Delta T_{\log}$ - logarithmic mean temperature difference, A – heat transfer surface.

Additionally, assuming that heat transfer in the considered case is primarily governed by flow velocities of both fluids, then simple relations for both series of investigations can be written:

For $\dot{m}_1$ = const. and $\dot{m}_2$ = var there is:

$$\alpha_1 = const \text{ , } \alpha_2 = C_2 w_2^{n_2} \tag{2}$$

For $\dot{m}_2$ = const. and $\dot{m}_1$ = var there is:

$$\alpha_2 = const \text{ , } \alpha_1 = C_1 w_1^{n1} \tag{3}$$

where α_1 and α_2 are heat transfer coefficients for respective mass flow rates; w_1 and w_2 are respective flow velocities; n_1 and n_2 are coefficients depending on the character of heat transfer, for example in case of turbulent flow inside tubes n=0.8, whereas in case of a laminar one, n=0.5.

Overall heat transfer coefficient determined from the heat balance yields:

$$k = \left(\frac{1}{\alpha_1} + \frac{\delta}{\lambda} + \frac{1}{\alpha_2}\right)^{-1} \tag{4}$$

where: δ – is a thickness of a wall separating two fluids, whereas λ its thermal conductivity.

Introducing the above relations for the first series of investigations we get:

$$\frac{1}{k}=\left(\frac{1}{\alpha_1}+\frac{\delta}{\lambda}\right)+C_2 w_2^{-n_2} \tag{5}$$

or: $$\frac{1}{k}=C_3+C_2 w_2^{-n_2} \tag{6}$$

where:

$$C_3=\frac{1}{\alpha_1}+\frac{\delta}{\lambda} \tag{7}$$

for a series where $\dot{m}_1$ = const. Assuming new variables, i.e. $x=w_2^{-n_2}$ and $y=1/k$ we obtain for the case of a first series of investigations a linear relation:

$$y = C_3 + C_2\, x \tag{8}$$

provided a value of coefficient n_2 is known. It can, however, in the course of data reduction be corrected until bet fitting of the results to the linear regression is obtained (Figure 9).

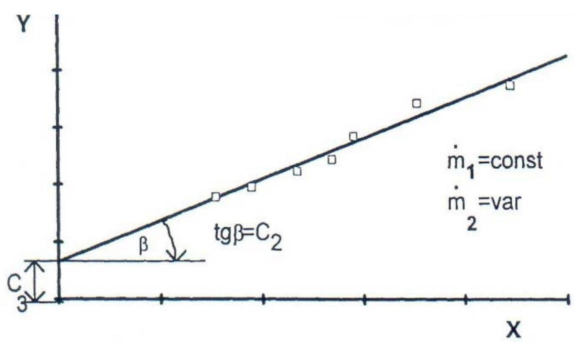

Figure 9 Example of determination of heat transfer coefficient using the Wilson method

Constants C_3 and C_2 are determined similarly as for the linear regression using the least squares method. Knowledge of these constants enables determination for the same series of investigations of a one value of heat transfer coefficient α_1 as well as a sequence of values for the heat transfer coefficient α_2. On the other hand, from a series of tests carried out at a constant flow rate $\dot{m}_2$ = const and varying flow rate $\dot{m}_1$ = var, further values of heat transfer coefficient α_1 can be determined. Also the mean wall temperature on the required side case can be determined from a relation:

$$\Delta T_{\log}=\frac{\left(T_{hot\,in}-T_{cold\,out}\right)-\left(T_{hot\,out}-T_{cold\,in}\right)}{\ln\dfrac{\left(T_{hot\,in}-T_{cold\,out}\right)}{\left(T_{hot\,out}-T_{cold\,in}\right)}}=\frac{\dot{Q}}{\alpha_1 A} \tag{9}$$

That is especially important in the case of finned tubes where determination of a mean value of wall temperature is difficult basing on local measurements.

The heat transfer coefficient calculations by Wilson's method were conducted for the plate thickness of 2 mm. The plate material (the aluminium alloy) has the thermal conductivity λ equal to 207 W/mK.

For the cold fluid the straight line was plotted and its equation was presented as eq. (8) **y = $\mathbf{C_3}$ + $\mathbf{C_2}$ x** (Figure 10), where constants have values of $C_2 = 92\times10^{-6}$ and $C_3 = 49\times10^{-6}$. For the hot fluid the plotted straight line (Figure 11) had the constants equal to $C_2 = 61\times10^{-6}$ and $C_3 = 60\times10^{-6}$.

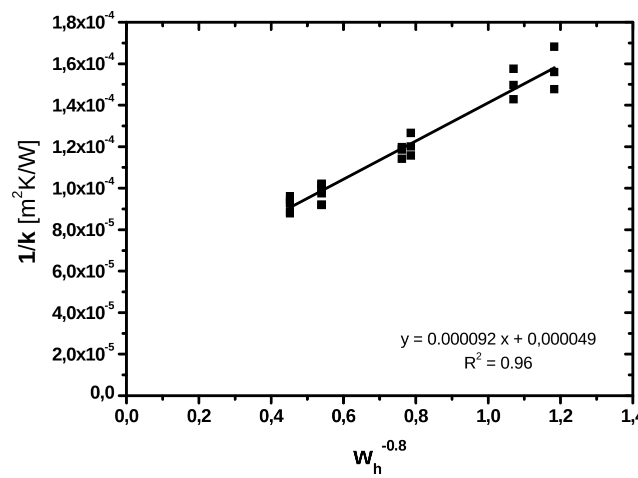

Figure 10 Experimental points and linear regression for $\dot{m}_c$ = 200 l/h

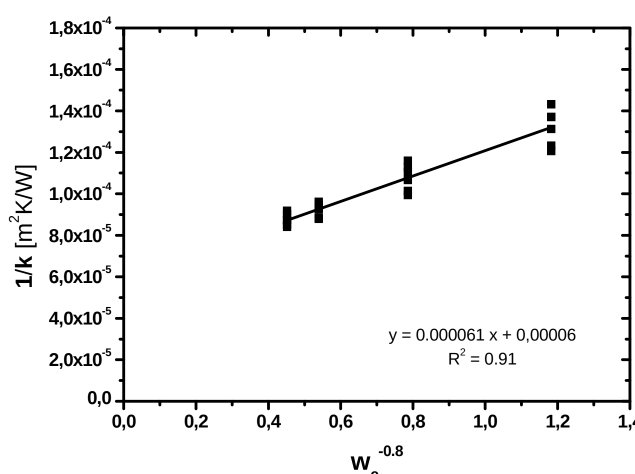

Figure 11 Experimental points and linear regression for $\dot{m}_h$ = 250 l/h

In Figure 12 the distribution of overall heat transfer coefficient of the heat exchanger in relation to the LMTD is presented. It can be seen that in the heat exchanger attainable is the overall heat transfer coefficient of the order of 12000 W/m^{2}K at LMTD of 60 K.

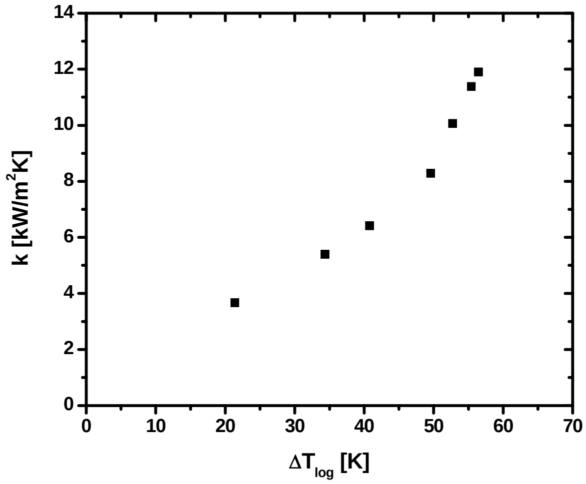

Figure 12 heat transfer coefficient versus LMTD

CONCLUSIONS

In the paper the original compact heat exchanger with microjets was proposed. Its primary application is for the domestic ORC. However, the design is universal and may find numerous applications. The technology of microjets manufacturing is a patented design by Plata [6]. In the present paper the idea of such heat exchanger was shown together with the flow and thermal characteristics of the prototype. The heat exchanger is capable of exchanging 5 kW of thermal energy at LMTD of 60 K. The obtained overall heat transfer coefficient reaches 12000 W/m^{2}K. Experimental data was also collected and the heat transfer coefficient was calculated using the Wilson method. The heat exchanger design will be further pursuit in the optimization with respect to the length of nozzles to reduce the pressure drop and increase heat transfer rates.

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